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REVIEW AND EXPERIMENTAL TEST OF THE STEAM PLANT OF THE S. S. *IROQUOIS.*

BY W. C. SELDEN, ESQ., MEMBER.

The accompanying review of the steam plant of the Clyde Line steamship *Iroquois* by Messrs. King, Powell and Shiebler, gives a very complete and accurate showing of the usual working conditions.

The engine is the usual triple-expansion type, with cylinders of 23, 36 and 60 inches diameter, and a stroke of 36 inches. The valves are a piston valve for the high pressure, and slide valves for the intermediate and low pressure, all actuated by Stephenson link motion.

The air, feed and bilge pumps are all worked in the usual manner by a rock-shaft from the low-pressure crosshead.

The condenser, which forms part of the engine frame, contains a cooling surface of 2,740 square feet, exclusive of the heads.

The vessel was the first to be built of steel in this country for the coasting trade, and was intended to carry passengers and a

large cargo on a light draught. While not expected to be remarkably fast, she has averaged over her course for the entire voyage a speed of 13 knots, but she is chiefly noted for her remarkably even running under all circumstances.

The following are the chief dimensions :

Length over all, feet and inches	292–0
Length between perpendiculars, feet and inches	280–0
Length from forward side of stem to after side of stern post, feet and inches	273–6
Beam extreme, feet and inches	46–0
Depth from top side of main deck, feet and inches	20–6
Mean draught during experiment, feet and inches	12–0
Area L.W.L. at given draught, square feet	10,119.
Area immersed midship section, square feet	474.3
Proportion of L.W.L. area to circumscribing parallelogram, per cent	80
Angle of entrance, 12-foot W.L., degrees	21
Angle of clearance, 12-foot W.L., degrees	41
Displacement at 12 feet mean draught, tons	2,850.
Admiralty coefficient	210.5
$L \times B \times D \times 0.75 \div 100$	1,835.4
$\frac{1}{2} B + D$	43.3
$L \div B$	6.1
$L \div D$	14.7

Review of the Steam Plant of the Steamer *Iroquois*, By H. D. King, W. B. Powell and A. Shiebler.

In the test described, we have attempted to determine the economy of the engines and boilers of the steamer *Iroquois*, of New York, during their operation under the ordinary practical conditions of traffic.

The steamer is one of the largest of the fleet belonging to the Clyde Line, carrying both freight and passengers between New York, Charleston and Jacksonville. She was built in 1888 by The Wm. Cramp & Sons Ship and Engine Building Co., at Philadelphia, and is 46 feet beam, 280 feet long between perpendiculars and 292 feet over all, and her tonnage is $2{,}943\frac{64}{100}$ gross, or $2{,}236\frac{3}{100}$ net register. Her draught is 12 feet to the load-water line, and she was designed to carry 265 crew and passengers,

and 151,000 cubic feet of freight. She is built practically without a keel, in order to pass over the bars at the entrance to Charleston harbor and the St. Johns River, but is furnished with bilge keelsons. Her propeller is 13 feet diameter, having four blades of 15 feet pitch, and a shaft of 10¾ inches diameter.

The motive power is supplied by a vertical triple-expansion engine of modern type with Stephenson's links. The air pump and the two feed pumps, as well as a sanitary and a bilge pump, are run by a rocker arm from the crosshead of the low-pressure cylinder. A plain slide-valve engine drives the centrifugal pump which circulates water in the condenser, while an independent Sweet "straight-line" engine is used five hours a day to run the dynamo for illuminating purposes. Aside from these, steam is also used in the steering engine, in the heater coils and in the galley, and occasionally in a duplex pump for general work. The steam for these uses is supplied by four horizontal return tubular boilers of the prevailing marine type, with two corrugated fire flues in each. These are arranged in plenum, and are kept in remarkably good condition, being cleaned almost every trip.

Under fair average conditions, the steamer maintains a speed of 12 to 14 knots an hour, and has covered the course from New York to Charleston, a distance of 630 miles, in a trifle over 48 hours.

Two distinct tests were made, the first on the way from New York to Charleston for 45 hours, and the second of 48 hours on the return. These tests will be treated separately throughout.

I. BOILERS.

The four boilers were equal in size and were arranged two on either side, with a common smoke stack 6 feet 6 inches in diameter and 52 feet 4 inches high, measured from the grate bars. The boilers were rated for 2,000 I.H.P., running with natural draft, and were tested to 240 pounds per square inch, but the safety valves were set for 150 pounds, and they seldom carry over 140. The dimensions of each boiler are as follows:

Length, feet	12
Diameter, feet	11
Number of fire flues	2
Diameter of fire flues, mean, inches	43
Grate in each flue, feet	$3\frac{1}{2} \times 6\frac{1}{2}$
Grate surface, square feet	22.75
Number of tubes	146
Diameter of tubes, inches	3
Length of tubes, feet and inches	8–$3\frac{1}{4}$
Heating surface, square feet	1,350
Combustion chamber, feet	$3 \times 9\frac{1}{2} \times 3$

From the above are found the following totals:

Boiler tubes	584
Grate surface, square feet	182
Heating surface, square feet	5,400
Ratio	1 to 29.4

FIRST TEST OF BOILERS.

In testing the boilers, all feed water was passed through a 3-inch Worthington steel water meter placed between the feed pumps and the boilers, the water in the glasses being brought to the same level at the beginning and the end of the test. The coal was roughly measured by shoveling into a box holding about 1,200 pounds, while the ashes were measured in buckets as they were sent up to be thrown overboard, each bucket weighing 100 pounds, on the average. In connection with the water meter, the temperature and pressure of the feed water were also taken. The difference in mean pressure between the pump side of the meter and the boiler was 4.17 pounds per square inch, due in part to bends in the pipe and the check valves, but mostly to the resistance of the water meter.

The total reading of the meter for the 45-hours test was 17,795 cubic feet. But when the meter was afterward tested by pumping 40 cubic feet of water into a standardized tank, a correction of 1.015 was found, which gave the actual water fed to the boiler 18,062 cubic feet. From this, we found that the meter passed 6.69 cubic feet of water per minute, which was at the velocity of 13.62 feet per minute.

The temperature of the feed water was raised in the hot well

by the exhaust of the circulating engine to an average of 163.15° Fahrenheit, as determined in a mercury well near the meter. The average boiler pressure for the run being 137.13 pounds per square inch, we found the factor of evaporation to be 1.097308, which gave the water evaporated from and at 212° Fahrenheit to be 1,207,404 pounds.

The coal used during the test filled 101 boxes, giving a total of 121,200 pounds. From this, the evaporation per pound of coal was found to be 9.96 pounds, which figure is quite high, but is undoubtedly quite accurate in view of the remarkably clean condition of the boilers and the good quality of coal used, which was considerably better than that often used on steamships. The ash and refuse filled 143 buckets, or a total of 14,300 pounds, leaving the amount of combustible 106,900 pounds. From this figure the evaporation per pound of combustible was found to be 11.29 pounds from and at 212° Fahrenheit. The quality of the coal is well shown by the low percentage of ash, *i. e.* 11.8 per cent.

From the dimensions previously given, the coal consumed per square foot of grate per hour was found to be 14.8 pounds. The evaporation per square foot of heating surface per hour at the actual pressure and temperature of the feed was 4.52 pounds, or 4.97 pounds from and at 212° Fahrenheit.

At the standard rating adopted by the American Society of Mechanical Engineers, the horse power of the boilers was 777.77, much less than that actually developed by the plant; but this was easily accounted for by the fact that the engines developed a horse power for less than 17, instead of 34½ pounds of water. It is quite probable that the boilers would develop more than their designed rating of 2,000 horse power if, in an emergency, they should be forced to their fullest capacity.

The results developed in the foregoing analysis are summarized in the following:

Coal consumed, pounds	121,200
Ashes and refuse, pounds	14,300
Combustible, pounds	106,900
Percentage of ash and refuse	11.8

Water evaporated, pounds:	
Total	1,100,332.5
Equivalent from and at 212°	1,207,404.
Per pound of coal	9.08
Per pound of combustible	10.29
From and at 212°, per pound of coal	9.96
From and at 212°, per pound of combustible	11.29
Per square foot of heating surface per hour	4.52
From and at 212°, per square foot of heating surface, per hour	4.97
Coal burned per square foot of grate, per hour	14.8
Rated I.H.P., A. S. M. E. standard	777.77
Heating surface per I.H.P., square feet	3.73
Same according to A. S. M. E. standard	7.08

The coal used during both tests was good quality of bituminous, coming from Cherry Creek, Pennsylvania; and its good composition, which accounts for the rather low percentage of ash and refuse above given, is shown by the following chemical analysis:

	1st sample, 2 grams.		2d sample, 1.977 grams.	
Moisture	.035 gr.	1.75 per cent.	.036 gr.	1.82 per cent.
Volatile matter	.360 "	18. "	.345 "	17.45 "
Carbon	1.512 "	75.6 "	1.502 "	75.97 "
Ash	.093 "	4.65 "	.094 "	4.76 "
	2.000 gr.	100.00 per cent.	1.977 gr.	100.00 per cent.

SECOND TEST OF THE BOILERS.

During the second test, all measurements of coal, water, and ashes were made in the same manner as previously, and all other conditions were held as nearly uniform as possible. The principal difference between the two tests was that, during the first, the ship rode perfectly even and steady, while during the second, she rolled considerably. This made no apparent difference in the steaming capacity of the boilers, many of the figures being almost identical. Whether a heavy pitching of the vessel would have any effect, we were unable to determine, as we had no bad weather during the entire trip, but it is doubtful if there would be any effect except a trifle due to the greater difficulty of firing.

The total reading of the water meter for this test, which lasted 48 hours, was 19,583 cubic feet, which, when corrected for the meter, became 19,876 cubic feet, giving the rate of passing the meter as 6.9 cubic feet per minute. The average temperature of the feed was 162.7° F., and the average boiler pressure 139.5 pounds per square inch, the difference between this latter and the average pressure on the pump side of the meter showing that only 5.5 pounds were required to pass the water through. From the pressure and temperature, we found the factor of evaporation to be 1.098033, from which the water evaporated from and at 212° F. was found to be 1,329,817 pounds.

The coal used filled the box 111 times, giving a total of 133,200 pounds, while the ashes and refuse weighed 18,200 pounds. From these figures it was found that the evaporation per pound of coal from and at 212° F. was 9.98 pounds. The percentage of ash here is a trifle higher than in the other test, being 13.66 per cent. instead of 11.8; but this difference is probably due to the greater difficulty in separating the coal from the ashes, and it is quite likely that some good fuel was thrown overboard and charged to the ash account.

The coal consumed per square foot of grate surface per hour was 15.26; the evaporation per square foot of heating surface per hour was 4.67 pounds at the actual pressure and temperature of feed, or 5.13 pounds from and at 212° F. The square feet of heating surface required for each horse power actually was 3.47, and the rating according to the Centennial standard was 803.3 H.P.

The figures of this test are summarized in the following table for convenience of comparison:

Coal consumed, pounds	133,200
Ashes and refuse, pounds	18,200
Combustible, pounds	115,000
Percentage of ash and refuse	13.66
Water evaporated, pounds:	
Total	1,211,090
Equivalent from and at 212°	1,329,817
Per pound of coal	9.09
Per pound of combustible	10.52

From and at 212° per pound of coal	9.98
From and at 212° per pound of combustible	11.56
Per square foot of heating surface per hour	4.67
From and at 212° per square foot of heating surface per hour	5.13
Coal burned per square foot of grate per hour, pounds	15.26
Rated I.H.P., A. S. M. E. standard	803.3
Heating surface per I.H.P., square foot	3.47
Same according to A. S. M. E. standard	6.72

This closes the figures determined during the tests of the boilers, which will be further discussed in connection with the engines. To illustrate how steadily the boilers performed their work, we append extracts from the logs for the two tests, in which one feature is particularly noticeable, namely, the gradual increase in temperature of the feed going down, and a corresponding decrease on the return, due no doubt to the differences in the temperature of the sea water circulating in the condenser.

Test No. 1.				Test No. 2.			
Time.	Pressure.	Water meter.	Temp. feed.	Time.	Pressure.	Water meter.	Temp. feed.
April 11.				April 18.			
8 P. M.	126	238,835	148.5	12 Noon	133	269.773	177
10	137	239,893	146.5	2 P. M.	140	270,480	174
12 Mid	132	240,489	156	4	142	271,207	174
April 12.				6	141	272.004	169
2 A. M.	139	241,240	158	8	141	272,837	170
4	140	242,027	165	10	131	273,637	172
6	139	242,829	161	12 Mid	143	274,408	176
8	138	243,646	165	April 19.			
10	138	244.427	160	2 A. M.	136	275,163	174
12 Noon	137	245.277	164	4	140	276,000	168
2 P. M.	139	246,048	165	6	140	276,834	172
4	138	246,862	166	8	141	277,655	175
6	138	247,669	163	10	140	278,496	169
8	137	248,480	167	12 Noon	140	279.343	165
10	143	249.326	171	2 P. M.	142	280,140	160
12 Mid	131	250,192	174	4	138	280.952	158
April 13.				6	139	281.725	155
2 A. M.	140	250,830	170	8	140	282.558	154
4	137	251.504	168	10	142	283,386	155
6	140	252,230	169	12 Mid	140	284,215	159
8	135	253.008	165	April 20.			
10	140	253,710	166	3 A. M.	135	285.445	152
12 Noon	133	254,550	168	6	141	286,620	154
2	140	255,276	169	9	141	287,839	151
5	135	256,680	171	12 Noon	143	289.356	153

II., ENGINES.

The engines were tested simultaneously with the boilers, and, as before, the two tests will be treated separately. Although we have already given a general outline of the distribution of the steam, we find a more detailed account of the steam plant needful in this connection, which may make a few repetitions unavoidable.

The main engines were of the upright, triple-expansion type, arranged longitudinally about amidships, and acting on cranks set at angles of 120°. The stroke of each piston was three feet, the diameters of the high, intermediate and low-pressure cylinders being respectively 23, 36 and 60 inches. The cut-off in the high-pressure cylinder occurred at $\frac{3}{4}$ of the stroke, and from there the steam was expanded to $9\frac{1}{5}$ times its volume. At the outset, indicator cards were taken at half hour intervals, but this frequency was soon found needless, and cards were taken hourly thereafter. Owing to the fact that the air, bilge, sanitary and feed pumps were all operated by a rocker arm directly from the low-pressure crosshead, no account of their individual performances was needed.

Aside from the main engines, however, steam was consumed for various other purposes which had to be considered. Principal among these was the small vertical slide-valve engine used to run the circulating pump, 6×6 inches, in which steam was throttled down from full boiler pressure. To determine the steam consumed and the power developed in this engine, it was indicated hourly. The straight-line engine which drove the dynamo was also indicated hourly during its five hours daily operation. This was a $12\times6\frac{3}{16}$-inch engine with 6 per cent. clearance, the exhaust being led into the condenser of the main engines.

Difficulties arose, however, with those consumers of steam which could not be indicated, such as the steering engine and the utility pump. In the case of the former, not only was there no means of applying an indicator, but there arose a greater difficulty in the fact that at one minute the engine was motionless, while in the next it was running at a speed of perhaps 200

revolutions. There remained nothing but the barest approximation, for which the only data taken were the reduced steam pressure and the total number of revolutions. The utility pump presented about the same difficulties, as it likewise was not tapped for indicators, and was run only at occasional intervals. Under these circumstances, the total length of time running, the average stroke and an assumed percentage of condensation furnished the basis for the nearest possible approximation. The only remaining consumption of steam was in the radiators and in the galley, from which places the condensed steam was returned to the hot well. No account of this item was taken during the first test, but on the second, the pipe was tapped where it entered the hot well, and the exhaust was all collected and weighed.

The vacuum for the low-pressure cylinder and the dynamo engine was obtained by exhausting into a water-tube surface condenser containing a total surface of 2,740 square feet. Through this condenser sea water was circulated by a 30-inch centrifugal pump driven as already described, the temperature being raised from 57° to 84°. From the condenser the condensed steam was returned to the hot well, at a temperature of 125°, being passed through a filter box on the way, to free it from oil. In the hot well, the temperature of the water was raised to 163° by the exhaust from the circulating engine, and from there it was sent to the feed pumps to be returned to the boilers.

Having a general outline of the engine plant now before us, we will proceed with the analysis of the tests.

FIRST TEST OF ENGINES.

During the first run of 45 hours, the engines performed their work with an average of 92¼ turns per minute. The average pressures available from the boilers and in the first and second receivers were, respectively, 137.13, 57.29 and 7.75 pounds per square inch, while the average vacuum was 25.97 inches of mercury. Under these conditions, the horse power developed as figured from the average of all the hourly indicator cards for the run was 1,448.67, distributed between the three cylinders as follows:

	I.H.P.
High pressure,	395.975
Intermediate pressure,	557.236
Low pressure,	495.46
Total,	1,448.671

During the run, however, owing to small differences in the conditions, the horse power varied considerably, the highest being 1,563, and the lowest 1,144, divided as follows:

High pressure,	397.335	332.59
Intermediate pressure,	613.454	443.659
Low pressure,	552.501	368.334
Total,	1,563.29	1,144.583

The average revolutions of the circulating engine were 172.8 per minute, but the reduced steam pressure under which it worked was not taken, as a gauge could not be applied. The horse power developed, as measured by the average of all the hourly cards, was 3.086.

The dynamo engine made two 5-hour runs during the test, in which the average revolutions were 279 per minute, and the reduced steam pressure 80 pounds per square inch. The average horse power developed in this case was 13.044, by the cards. The engine ran with remarkable regularity under variations of load, being regulated by an automatic fly-wheel governor.

The accompanying cards from the different engines are submitted as samples, they having as nearly as possible the same mean effective pressure as the average for the entire run. It will be noticed that the cards from the two ends of the circulating engine are entirely dissimilar, and, to say the least, peculiar. After considerable study, we decided that the principal fault was that the inside lap at one end of the valve and the outside lap at the other were too large, and an examination proved that our conclusions were well founded.

(NOTE.—It was discovered, some time after this paper was written, that a flaw in the casting of the steam passages allowed steam to leak directly through from the steam port of one end of the cylinder to the exhaust port, thereby fully explaining the peculiar card from one end of the cylinder.)

The total water per indicated horse power per hour, making no allowances for the water used for auxiliary purposes, was 16.8 pounds, equivalent to 1.86 pounds of coal per hour. When, however, deductions are made for the steam consumed in other ways, these figures will be somewhat reduced, and these deductions we will now consider.

Circulating Engine.—Calculation of the water from indicator card.

In the circulating engine, the clearance of the piston was $\frac{1}{4}$ of an inch, which, together with the volume of the ports, determined the clearance to be 6 per cent. The revolutions were 184 per minute, and the cylinder condensation was assumed at 25 per cent., as the cylinder was small and not jacketed. From these figures, the water per revolution was found to be .01667 pounds from the card given as a sample. This gives the total water used by the engine in 45 hours to be 11,042.2 pounds, or 59.6 pounds per I.H.P. per hour, which may be a trifle above the usual performance of such engines, but which is easily accounted for by the irregularity of the operation of the engine.

Dynamo Engine.—Calculation of the water from indicator card.

In this case, the piston clearance was $\frac{5}{16}$ inch, to which was added the volume of the ports, making the total clearance 6 per cent. The revolutions were 286 per minute, and the cylinder condensation was assumed at 20 per cent., the engine being high speed without a jacket. From the sample card given, the water per revolution was calculated as .015064 pounds, giving a total for the two 5-hour runs of 3,236.2 pounds. This is equivalent to 24.85 pounds of water per I.H.P. per hour, which may be a trifle low, caused probably by the short cut-off and the fact of its exhausting into the condenser.

Steering Engine.—Approximation, by assumptions, of water used.

In the case of the steering engine, there were two cylinders, each 5 inches diameter and 6 inches stroke, which made 59,770 revolutions during the test. The average steam pressure was 32 pounds per square inch, which was available throughout the stroke, as there is no cut-off in this class of engines. Owing to the irregularity of action, 50 per cent. was considered a safe figure for the cylinder condensation. From these figures the water per revolution was determined to be .0598224 pounds, which gives 3,575.58 pounds for the entire run.

Donkey Pump.—Approximation, by assumptions, of water used.

This utility pump was a Blake duplex, 12 inches diameter and 10 stroke, and was run in all 131 minutes to wet down the ashes. It ran under about 50 pounds per square inch pressure without cut-off, making an average of 25 strokes per minute. The cylinder condensation was assumed at 60 per cent. on account of its slow operation. From these figures the water per revolution was found to be nearly one pound, equal to 3,250 pounds for the whole time.

Heater Coils.—During the test, no account was taken of the steam used in these devices, and so no substantial allowance can be made. We will omit it in this case, and consider it only in the second test, it being insignificant at best.

Summary.—Having made allowance for the steam used in all the various auxiliary appliances, the steam actually used by the main engines was found to be 1,079,228.5 pounds, as follows:

Circulating engine, pounds,	11,042.2
Dynamo engine, pounds,	3,236.2
Steering engine, pounds,	3,575.58
Donkey pump,	3,250.
Total allowances, pounds,	21,104.
Water actually evaporated, pounds,	1,100,332.5
Water used by main engines, pounds,	1,079,228.5

From this figure the water per indicated horse power per hour was found to be 16.5 pounds, an equivalent of 1.81 pounds of coal per hour.

SECOND TEST OF ENGINES.

It may, perhaps, save needless repetition to state at the outset of this discussion that this test was conducted exactly as the previous one was, and that throughout the calculations the same assumptions have been made and the same methods of deduction followed.

The average indicated horse power during this test of 48 hours was larger than before, being 1,557, distributed according to *a* in the following table. The average pressures in the boiler, and the first and second receivers were, respectively, 139.5, 59.77, and 8.23 pounds per square inch, while the average vacuum was 25.76 inches of mercury. The average revolutions were almost identical, being in this case 92.4. The highest horse power developed during the run was 1,673, while the lowest was 1,401, as follows:

	a	*b*	*c*
High pressure, ...	421.675	456.388	369.915
Intermediate pressure, . .	593.339	633.629	548.092
Low pressure, ...	542.174	583.196	482.993
Total, ...	1,557.188	1,673.213	1,401.

The average revolutions of the circulating pump were 205.17, which produced a horse power of 4.638, more work being required on account of the higher temperature of the sea water. During the two runs of the dynamo engine, the average pressure was a trifle higher, being 83.6 pounds per square inch, while the revolutions were 276 per minute. The average horse power developed was 12.207. The cards submitted are samples, having been selected, as before, for their near approach to the aver-

ages. The same curious features will be noticed in the cards of the circulating engine. While the steamer was in port at Jacksonville, an attempt had been made, to remedy this defect by re-setting the valve, but the cards taken on this run showed that something else was the matter, as we have already explained.

During this test, the water per indicated horse power per hour without allowances was 16.2 pounds, equivalent to 1.78 pounds of coal. The allowances for the auxiliary engines were figured as before, with the following results.

Circulating Engine.—Calculation of the water from indicator card.

The water per revolution was .01859 pounds, and the revolutions being 214, the total amount of water used, allowing 25 per cent. condensation, was 15,276.5 during the test.

Dynamo Engine.—Calculation of the water from indicator card.

In the case of our sample card for this engine, the revolutions were 277 per minute, and the water per revolution was .0156 pounds, giving a total of 3,240.9 pounds for the test with 20 per cent. condensation.

Steering Engine.—Approximation, by assumptions, of water used.

The total number of revolutions made during this test was 97,957, and the average steam pressure 43 pounds per square inch, from which the total water was calculated as 7,137.68 pounds.

Donkey Pump.—Approximation, by assumptions, of water used.

The donkey during this test was only run for 53 minutes, which gives the total water consumed in this way as only 1,324.22 pounds.

Heater Coils.—Water used, as actually measured.

During this test all the steam used in heating the ship was drawn off into a large can, instead of being run into the hot well, and was thereby measured. This can full of water weighed 85 pounds net, and was filled 30½ times during the run, giving a total of 2,592.5 pounds of water.

Summary.—The aggregate of all these allowances gives the actual water used by the main engine as follows:

Circulating engine, pounds, . . .	15,276.5
Dynamo engine, pounds, . . .	3,240.9
Steering engine, pounds, . . .	7,137.68
Heater coils, pounds, ...	2,592.5
Donkey pump, pounds, ...	1,324.22
Total allowances, pounds, . .	29,571.8
Water actually evaporated, pounds, .	1,211,090.
Water actually used by main engine, pounds, ...	1,181,518.2

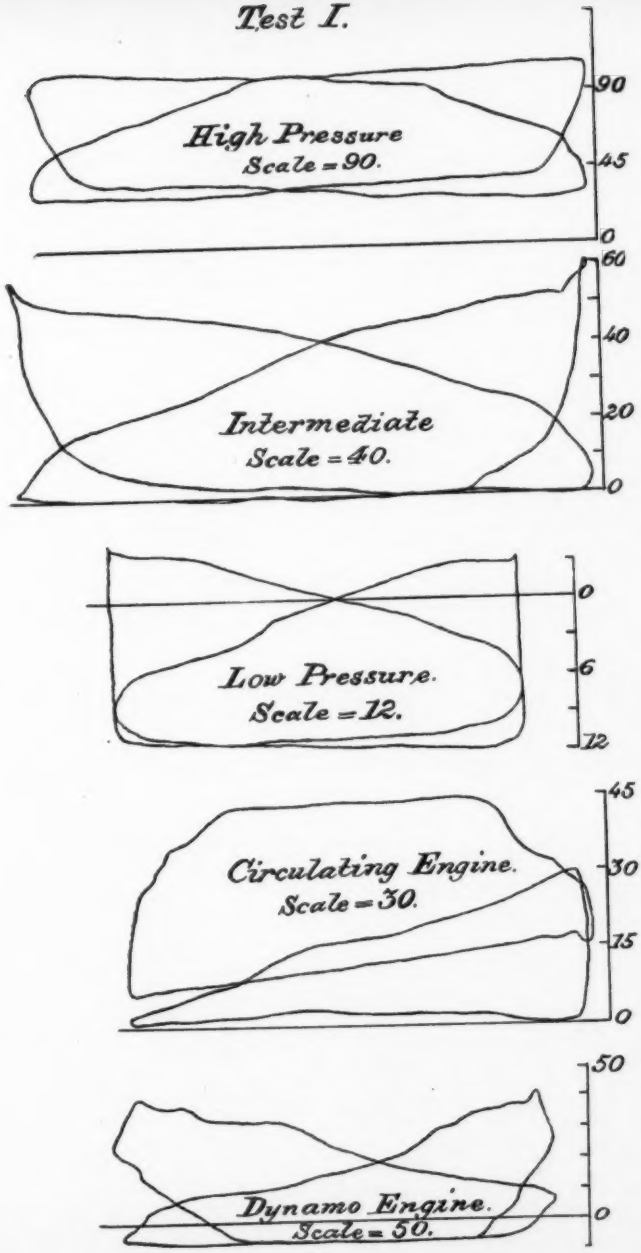
From this figure the water per indicated horse power per hour was found to be 15.8 pounds, equal to 1.74 pounds of coal.

III. EFFICIENCIES.

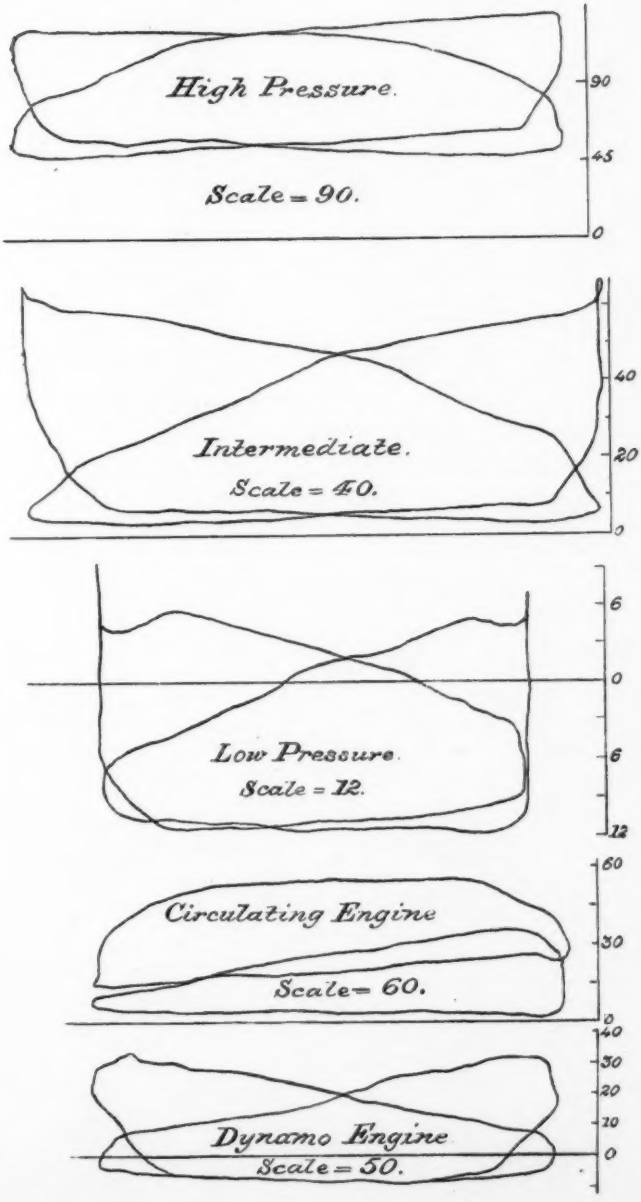
For the determination of the heat units contained in the Cherry Creek coal which was burned during the foregoing tests, we are indebted to Messrs. Waefelaer and Vidal, who kindly analyzed a couple of samples with Dr. Ferdinand Fischer's calorimeter. The following figures are given as they were furnished to us by these gentlemen:

CO_2 from CO, ...	2. mg.	None.
Free hydrogen, ...	1.2 mg.	2.2 mg.
Condensed water, . . .	198. mg.	103. mg.
Coke, ...	16. mg.	25. mg.
Initial temperature, . . .	21.35° C.	20.55° C.
Final temperature, . . .	26.1° C.	25.25° C.
Difference, ...	4.75° C.	4.7° C.
Ash, per cent., ...	3.95	4.
Total heat units, B.T.U., .	13,592.07	13,744.89
Pounds of water which would be evaporated from and at 212°, ...	14.07	14.22

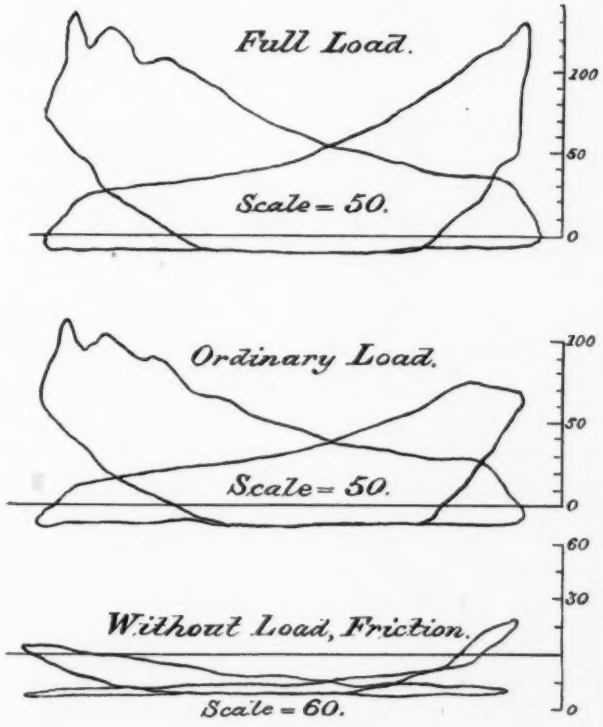
Test I.


90
45
0
High Pressure
Scale = 90.
60
40
20
0
Intermediate
Scale = 40.
0
6
12
Low Pressure.
Scale = 12.
45
30
15
0
Circulating Engine.
Scale = 30.
50
0
Dynamo Engine.
Scale = 50.

Test 2.


High Pressure.
Scale = 90.
90
45
0
Intermediate.
Scale = 40.
40
20
0
Low Pressure.
Scale = 12.
6
0
6
12
Circulating Engine
Scale = 60.
60
30
0
Dynamo Engine
Scale = 50.
40
30
20
10
0

Dynamo Engine Test.



Or taking an average of the results for the two samples, they find that the coal contained 13,668.48 B.T.U., which, if all utilized, would evaporate 14.14 pounds of water from and at 212°.

From the foregoing figures we may now compute the efficiency of the boilers, by comparing the actual evaporation from and at 212°, which was 9.97 pounds for the two tests, with the water which would have been evaporated if all the heat in the coal had been utilized. From this the efficiency of the boilers was found to be 70.5 per cent.

The efficiency of the plant (engines and boilers combined) is the thermal equivalent of the work done by one horse power in one hour divided by the heat of combustion of the coal burned per hour per indicated horse power. The former quantity as usually employed by engineers is 2,564.7, being derived on the basis of 772 foot pounds per heat unit. For the latter quantity, we have already determined the coal burned per hour per horse power in each test, which, when multiplied by the heat units per pound, gives the figure required. The efficiency of the plant was determined to be 10.3 per cent. in the first test, and 10.7 in the second; or an average efficiency of 10.5 per cent.

Dividing this latter figure by the efficiency of the boilers, the efficiency of the steam is found to be 14.6 per cent. and 15.7 per cent. in the two tests respectively.

Unfortunately, the efficiency of the engines alone could not be determined, as there was no available means of running the engines without load, so as to determine the power absorbed by friction. We regret that we are, therefore, forced to conclude this analysis without that figure.

Under the heading of efficiency, the slip of the screw might also be placed, although it is hardly to be termed an efficiency. Since the pitch of the propeller was 15 feet, the revolutions for 24 hours during the first test showed a distance traversed of 328 nautical miles. During the same time the log showed an actual distance of 300 miles, from which the slip was determined as 8½ per cent. During a 27-hour run on the return test, the apparent slip was found to be 19 per cent., the increase being due to head winds and possibly to disadvantageous currents.

We have also to regret that our inability to secure or accurately determine the clearances of the main engine, has prevented us from properly concluding the analysis of the engines by presenting some compound cards with the interesting deductions arising therefrom.

IV. ELECTRIC PLANT.

The electric plant for the illumination of the vessel consisted of a Weston 200-light, shunt-wound incandescent dynamo, intended to develop 110 volts, which was driven by the Sweet straight-line engine already mentioned. Under ordinary circumstances, about 162 lamps were in use, requiring .44 amperes each. The current varied from 68 to 76 amperes, while the voltage ranged from 92 to 95½. Doubtless this drop in potential below the rating given above was due to the fact that the dynamo was not run up to speed, the revolutions being about 850 instead of the requisite 1,200 per minute. Under a load of 125 lamps the voltage rose to 110. The plant was operated five hours daily, and was separately tested during the return run from Jacksonville to Charleston.

Readings of the revolutions of the engine, amperes, and voltage were taken every 15 minutes, together with cards from the engine. The results are given in the following log:

Time.	Revs.	Amperes.	Volts.	Time.	Revs.	Amperes.	Volts.
6.45	280	70	95.	8.45	274	73.	95.
7.00	276	70	94.5	9.00	270	73.	94.
7.15	274	70	94.	9.15	284	70.5	95.
7.30	268	71	93.	9.30	266	70.	95.
7.45	270	73	92.	9.45	280	69.5	95.
8.00	278	72	92.	10.00	284	70.	94.5
8.15	276	76	95.	10.15	270	68.5	95.5
8.30	268	76	94.				
				Average.....	274.53	71.5	94.23

The engine ran under a constant steam pressure of 87 pounds per square inch, and developed 11.466 horse power figured from the average of all the cards. The friction of the engine and

dynamo was ascertained, by taking cards with the circuit broken, to be 1.818 horse power. Owing to the inaccessibility of the mechanism, the dynamo could not be cut off so as to ascertain the friction of the engine alone.

From the figures given by the above averages, the product of the volts and amperes gave 6,737.445 watts, or 9.031 electrical horse power. The net horse power, excluding friction, being 9.648, there remains a loss of .617 horse power due to hysteresis, etc. The efficiency of the mechanism was found, therefore, to be 78.7 per cent.

NOTE.—The drop in potential due to the dynamo not turning up to speed was accounted for, in part at least, by the slip of the belt from the engine to the dynamo. This was determined by taking simultaneous readings of the revolutions of each, the averages of which were 270 and 844.23 for the engine and dynamo respectively. The pulleys were 52 and 16 inches in diameter, from which figures the slip of the belt was determined to be 3.92 per cent.

Conclusion.—The plant, as a whole, is well adapted to the work it has to perform, and each individual part works with fairly good economy. A few minor details might be altered to increase the economy, but the improvements would be but a trifle compared with the whole. The improvement of the circulating engine, the defect in which we have already pointed out, has been effected.

We had hoped to obtain the theoretical horse power of the boat, but we regret that we have been unable to obtain either the lines or the augmented surface.

In conclusion, we desire to express our thanks to the engineer of the line, the chief engineer of the steamer, and to the general managers of the company for their kindness and courtesy in affording us the opportunity to make the test, and in aiding in every way in our work.

THE QUALITIES AND PERFORMANCES OF RECENT FIRST-CLASS BATTLESHIPS.

By Mr. W. H. White, C. B., LL. D., F. R. S.

[Read before the Institution of Naval Architects, 1894.]

Five years ago I read before this Institution a paper "On the Designs for the New Battleships," which were then about to be laid down under the provisions of the Naval Defense Act. Considerable public interest was shown in that paper, and in the discussion which followed, as was natural under the circumstances. Eight ships were to be built exceeding in dimensions any vessels previously constructed for the Royal Navy, and there were differences of opinion respecting the features of offense and defense embodied in the designs, as well as the policy of building vessels of such large dimensions and cost.

It is not my intention in the present communication to revive past controversies. The ships have been built, five of them are already in commission, one has been on active service since May, 1892, and the remaining three will be ready for service this month, when the five years end which were contemplated in the Act of 1889. My wish is to place before the Institution certain facts of professional interest in relation to the qualities and performances of the completed ships, and incidentally to compare the intentions of the designs with the results obtained in the finished ships.

Seven of the ships are of the barbette type, with high freeboard throughout the length, carrying their 67-ton guns about 23 feet above water. The *Royal Sovereign* is the typical vessel in this group which will be most frequently mentioned, as she was first completed, and has been nearly two years in commission. Two others of the class were built in the dockyards, and four by private firms. The machinery for the three dockyard

ships was supplied by one firm. For the contract-built ships the firms supplied machinery as well as ships. One design and specification served for the seven vessels, so far as hull and fittings were concerned. For propelling machinery there were four designs, all practically governed by the same conditions for weight, space and development of power on the contractors' trials.

The eighth ship (the *Hood*) is of the turret type, with moderate freeboard at the ends, over which the heavy guns deliver their fire. Those guns are about 17 feet above water. In the remaining features the *Hood* resembles the barbette ships, and her machinery was supplied by the same firm which undertook the corresponding work for the other dock-yard built ships.

Without further preface I will pass on to a statement of facts in regard to the ships as completed.

1. *Draught and Trim.*—The designed mean draught, with all authorized weights, and 900 tons of coal on board, was 27 feet 6 inches; the designed trim was 1 foot by the stern. A "Board margin," or unappropriated weight, of about 500 tons was included in the load, to be carried at the designed draught. During construction various additions were made to the hull, equipment and machinery, amounting in the aggregate to about 250 tons, the corresponding weight, with Board approval, being taken out of the margin.

When the *Royal Sovereign* was completed, a careful investigation was made, and it was found that at the designed mean draught she could carry 1,100 tons of coal, with all other authorized weights on board. In other words, the intentions of the design were fully realized, and 1,100 tons of coal could be carried instead of 900 tons, on the specified draught. The ship in this condition was also to her designed trim.

Similar results have been obtained in the sister ships built in the dockyards. As might be anticipated, there are some differences in weights between the sister ships built in different yards, and engined by different firms, but in no case is there any departure worth mentioning from the intentions of the design as regards draught and trim.

This is very satisfactory, having regard to the great sizes of the ships, and their many novel features. In the completed ships, certain additions have been made to the complements and consumable stores, as originally proposed. For example, the coal supply at normal draught was first fixed at 900 tons, with a bunker capacity of about 1,100 tons. The "wing spaces" abreast the bunkers proper were so constructed that they could be used as "reserve bunkers" if occasion arose, but it was not then contemplated to fill these spaces ordinarily. After experience in the *Royal Sovereign*, it has been decided to carry the full weight of coal when leaving port, so that about 1,450 tons of coal are available. Of course, the draught at starting is correspondingly increased.

Another alteration in practice subsequent to the design is the introduction of "reserves of feed water" for the boilers. Certain compartments in the double bottom have been utilized for the purpose, and from 130 tons to 160 tons of fresh water are carried at starting. In addition, the vessels have powerful evaporating plant.

For each additional inch immersion in these ships at load draught the increased displacement is about 55 tons, so that the extra coal and fresh water involves about 1 foot increase in draught. With their considerable freeboard and reserve of buoyancy this is of no importance, and in their fully laden condition, with bunkers full, the defensive powers are greatly increased.

2. *Stability: Metacentric Height.*—The determination of appropriate conditions of stability for these large ships was a matter of the most anxious consideration during the preparation of the designs.

One had to deal with circumstances in many ways unprecedented, particularly in regard to the loads of armor and armament, and the heights of the heavy guns and their protecting barbettes. It was recognized that these weights and their distribution would give the ships unusually large "moments of inertia," thus tending to long periods of oscillation, even if a good "metacentric height" was provided.

In regard to the metacentric height, it had to be kept in view that provision must be made, not merely for possible variations in stiffness occurring on service, and for sufficient "stiffness" when rapidly manœuvring, but for reductions in stiffness arising from damage in action. Here entered the consideration of the system of hull armor adopted, which gave a much greater protected area on the broadside—including the water-line belt and the lightly armored citadel standing on the belt deck—than had been possessed by many preceding classes in the Royal Navy, or was proposed for the most recent battle-ships in some foreign fleets.

An unduly large metacentric height was undesirable, of course, tending to lessen the periods of oscillation, and to produce unsteadiness of platform under ordinary conditions at sea. A small metacentric height, while it would have lengthened the period and conduced to steadiness, might have involved a lack of stiffness for rapid manœuvring of the ships, or for training their heavy guns on one broadside. If reduction in metacentric height had been carried too far, there would have been an undue reduction in the margin for possible damage in action. In existing ships examples are to be found of the evils attaching to either of these extremes in metacentric heights.

Moreover, I had to deal with an "unappropriated weight" of over 500 tons, and it was quite uncertain in what vertical position this might finally be placed, but judging by past experience the probability was it would be added in upper weights of additional armor or armament, boats and equipment.

Finally, I decided to aim at a metacentric height of about $3\frac{1}{2}$ feet for the barbette ships when in normal load condition at the mean draught of $27\frac{1}{2}$ feet with 900 tons of coal on board. Past experience had shown that in ships of much less size and inertia, an excellent combination of stiffness and steadiness had been obtained by the adoption of metacentric heights varying from $2\frac{3}{4}$ feet to $3\frac{1}{2}$ feet. In the new ships, having regard to their greater size and inertia, it was obvious that with a metacentric height of $3\frac{1}{2}$ feet, the period of oscillation should be at least as great as, and would possibly exceed, the corresponding periods for their

predecessors whose reputation for steadiness was so great. The *Hercules*, *Sultan*, *Inconstant*, *Monarch* and *Invincible* were amongst the vessels considered, as well as some foreign ships with equal reputations for steadiness at sea.

The loads of consumable stores in the new ships were so considerable that special care was necessary in disposing them, so that the variations in the trim and stiffness produced by the consumption of these stores might be minimized. The range of draught between the deep-load condition and the ordinary light condition is 3 feet, corresponding to a change of nearly 2,000 tons in displacement. By suitable arrangements it was made possible for the ships to pass from the normal condition to the extreme light condition, with practically no change in metacentric height.

When the full weight of coal (1,450 tons) is carried, since the reserve wing bunkers are low down, the metacentric height is increased to about 3¾ feet. The addition of the reserve feed in the lower compartments of the double bottom (made, as explained above, subsequently to the date of the design) involves a further increase in stiffness, bringing the metacentric height up to 4 feet, when wing bunkers, as well as ordinary bunkers, are full. This maximum stiffness may be decreased, of course, as coal and fresh water are consumed. It is interesting to add that it equals the metacentric height of the *Minotaur* class, which have good reputations for behavior at sea, and are of much less inertia.

The practical constancy in stiffness obtained in the *Royal Sovereign* class, apart from reserve bunkers and feed water, is in marked contrast to what was formerly common in armored ships where the coal was carried low down. For example, in the *Hercules* the metacentric height varied from 1.9 feet when light, to 2.7 feet when fully laden, and in the *Devastation* the corresponding variation was from 2.5 feet to 3.7 feet.

Inclining experiments were made on the *Royal Sovereign* and the *Ramillies* (contract built) to check the design calculations. In the *Royal Sovereign* the actual position of the center of gravity was found to be .15 feet (1¾ inch) below the calculated position. In the *Ramillies*, owing to heavier machinery, it was

about 4 inches below the calculated position. This close agreement between estimated and actual position was most satisfactory, and it was decided to be unnecessary to incline any more of the barbette ships. The turret ship *Hood* was also inclined with satisfactory results.

Members of the Institution will understand that the satisfactory solution of this problem was not an easy matter. On all sides there were limiting conditions. The features of armor, armament, freeboard and height of guns decided upon for the class, practically regulated the distribution of the principal weights in a vertical sense, and governed the vertical position of the center of gravity. The dimensions and forms of existing docks had to be considered, beam and draught could not be permitted to exceed certain maximum values. For manœuvring purposes the length had to be kept within reasonable limits. Stability had to be secured in association with a form adapted for high speed. This, I am happy to say, has been accomplished.

Some critics of warship designs entirely fail to recognize the part which stability plays in the selection of forms and proportions. They do not appreciate the radically different vertical distribution of weights in warships as compared with merchant ships. Consequently they suggest the adoption of cross-sectional forms and proportions of beam to draught and depth which would utterly fail to provide sufficient stiffness in warships. Should effect ever be given to their views in actual ships, the result would be disastrous. There is, however, no risk in that direction, for closer scrutiny and exact calculation would disclose the danger.

There is usually behind these suggestions some idea of possible advantage to be gained in a particular direction. For example, a favorite view is that a flatter floor, sharper bilge, and nearly vertical side above the bilge, would give greater displacement, in proportion to wetted surface, than the rounder bilge usual in warships. This is quite true. Further, I have seen it stated that the alternative form would increase resistance to rolling, and give greater steadiness than the actual sectional form of our battleships. This is a matter of opinion. I will not enter

upon its discussion now, but will only remark that, in my judgment, an altogether exaggerated value is attached in some quarters to the influence of sectional form in the central portions of the length upon rolling. One of the roundest forms of midship section ever adopted is that of the *Sultan*, and she has proved herself one of the steadiest ships afloat. Moreover, ease of rolling motion is a matter that must not be ignored, and in that respect the rounder form has distinct advantages. What is most important to note, however, is that such changes in cross-sectional form lower the metacenter and diminish stability.

For purposes of information only, Fig. 1 has been drawn, showing the comparative forms of midship section in the *Royal Sovereign*, a large ocean steamer of high speed, and the *Sultan*.

3. *Curves of Statical Stability.*—In Fig. 2 are given the curves of stability for the *Royal Sovereign* in various conditions. They are of an entirely satisfactory character. The maximum righting moment occurs at an inclination to the vertical of 37 degrees; the maximum value of the arm of the righting couple is $2\frac{1}{3}$ feet, and the range of stability is over 60 degrees.

Apart from the heave and impulse of the sea, the only inclining forces to which these ships, which have no sails, can be subjected are those when the vessels are turning rapidly, and those arising from wind pressure. Neither are of any practical importance. Turning trials show the ships to heel very little. If a wind having a velocity of 75 miles an hour acted directly on the whole broadside, the angle of steady heel would not exceed 4 degrees, and would probably be less.

Figs. 3 and 4 contain curves of stability for two typical ships which have had long and successful experience at sea. These afford an interesting comparison with Fig. 2.

Fig. 3 shows the curves for the *Monarch* as she has hitherto served, with good sail power supplementing steam. It is interesting to note the diminution of stability in the light condition. Her sail equipment, of course, involved much more serious demands upon her stability than would have been made had sails been absent. But that matter was so fully discussed for the

Monarch after the loss of the *Captain* that further remarks are unnecessary.

Fig. 4 shows the corresponding curves for the *Devastation* class of the Royal Navy. These vessels have no sails. Here the moderate freeboard, when fully laden, tends to lessened stability. As the ship lightens, although there is a loss of metacentric height, the gain of freeboard nearly counterbalances it.

Fig. 5 has been constructed for the load conditions of the *Royal Sovereign*, *Monarch* and *Devastation*, the ordinates representing righting moments (not arms of righting couples). Here the greater weight of the *Royal Sovereign* tells as compared with the other two ships.

Fig. 6 has been constructed for the light conditions of the three ships, also to show righting moments.

As against external forces, such as wind pressure, Figs. 5 to 6 are useful means of comparison. Under wave impulse Figs. 2 and 4 are to be preferred.

4. *Period of Oscillation.*—No rolling experiments in still water have yet been made on the *Royal Sovereign* class. From observations at sea, however, it appears that the period for a complete oscillation (double swing) is from 15 seconds to 16 seconds. The time for a single swing—port to starboard, or *vice versa*—is $7\frac{1}{2}$ seconds to 8 seconds. Variations in stiffness due to alterations in lading, of course, affect the period; but with normal weights, and $3\frac{1}{2}$ feet metacentric height, the period of oscillation is about 8 seconds. As above stated, this result was anticipated at the time the designs were prepared, and it brings the *Royal Sovereign* class, so far as period is concerned, into practical agreement with a number of ships which have great reputations for steadiness. Most of these ships have smaller metacentric heights than the *Royal Sovereign*, but they are also inferior in weight and moment of inertia.

Some persons, in discussing the behavior of the *Royal Sovereign* class, have dwelt upon their relatively large metacentric heights, but failed to remark their great moments of inertia. Both are counted into the period of oscillation, and that is the

best standard of comparison. The reasons for the selection of the metacentric height actually possessed by the *Royal Sovereign* class have been stated above.

5. *Bilge Keels.*—In this connection reference may be made to the reasons which led to the decision not to fit bilge keels to the *Royal Sovereign* class during construction. No one has more strenuously urged the utility of bilge keels than myself. But while it is undoubted that they can never do harm, and in many cases may do great good in limiting rolling, their influence varies under different circumstances. In small ships of quick period, and in ships of moderate weight and moment of inertia, they are likely to prove most beneficial. In slow-moving ships of great inertia the influence of bilge keels must clearly be less felt, and may be very small. That influence depends upon the area of the keels, their radial distance from the axis of rotation, and their rate of motion through the water, which rate is governed by the arc of oscillation and the period of the oscillation. Since, within considerable limits of oscillation, ships are practically isochronous in their motions, the mean rate of motion of bilge keels through the water varies with the arc of oscillation. In other words, fluid resistance to the motion of the bilge keels has a small value and moment about the axis of rotation in a slow-moving ship until considerable angles of oscillation are reached.

Moreover, with a given moment about the axis of rotation of fluid resistance to the motion of bilge keels, the check put upon the rolling of a ship to which they are attached depends upon the weight, the stiffness, and the moment of inertia of the ship. It may happen, therefore, in large slow-moving ships that the influence of the largest practicable bilge keels will be scarcely felt until considerable angles of rolling are reached.

This appeared to be the case in the *Royal Sovereign* class, and it was, therefore, decided not to fit bilge keels at the outset, until experience had been gained at sea with the ships. In thus deciding, the fact was kept in view that two vessels of exceptional steadiness, the *Hercules* and the *Sultan*, were practically without bilge keels, the side keels being very shallow in depth. More-

over, it was obvious that if bilge keels could be omitted, it would greatly facilitate docking the ships.

As an experiment, bilge keels have now been fitted to the *Repulse*, and she will be tried in company with other ships of the class. Provided that these trials are made under the same conditions of lading in all the ships, the results will be very valuable. Should the keels prove sensibly advantageous, they can easily be added to the sister ships.

It is unnecessary to repeat what has been said above with reference to the supposed influence of cross-sectional forms upon resistance to rolling. The analysis of a large number of rolling experiments on ships of very different cross-sectional form has shown that the portion of the resistance contributed by the midship parts of the length is small when compared with those portions of the resistance due to the deadwoods and flat surfaces, while surface disturbance greatly exceeds both of these factors.

One remark may be added. In recent years the battleships of the Royal Navy have, for the most part, been constructed with low ends. Hence moderate angles of rolling immerse the edges of the low decks and greatly increase resistance to rolling. There can be no question but that this deck resistance is a valuable feature in limiting oscillation under many circumstances.

In high-freeboard vessels of the *Royal Sovereign* class there is no corresponding element of resistance to rolling. On the other hand, the high freeboard and great height of guns increases fighting efficiency, the power of maintaining speed in rough water, and comfort to all on board. Under circumstances when the low-freeboard ships have to be battened down, or when they can fight their guns with difficulty, the high-freeboard ships suffer from no similar drawbacks.

6. *Behavior at Sea.*—Experience at sea with the new battleships has been very limited, except in the case of the *Royal Sovereign*, which was commissioned in May, 1892, as flagship of the Channel Squadron. The sister ships, *Empress of India* and *Resolution*, were commissioned for service with that squadron in September and December last. The turret ship *Hood* joined the

Mediterranean Fleet in June, 1893, and the *Ramillies* became the flagship on that station in October; but except on the passage out these ships have not had experience in the Atlantic. It is to the *Royal Sovereign*, therefore, that one must turn for information as to behavior over any considerable period. Apart from long-continued experience under various conditions of wind and sea, no fair appreciation is possible of the true qualities of any ship or class. I propose, therefore, to briefly summarize the facts for the *Royal Sovereign*, and subsequently to deal with the case of the *Resolution*, which was so much before the public in January last.

It will be convenient here to recapitulate the broad conclusions in regard to the rolling of ships, which have been established by theoretical investigation, and confirmed by experiment and observation during the thirty-four years that have elapsed since the late Mr. W. Froude brought his new theory before this Institution.

1. The behavior of a ship depends chiefly upon the ratio of her period of oscillation to the apparent period of the waves which produce rolling.

2. The slope of the waves, their ratio of height to length, is an important factor in the rate of accumulation of rolling and its maximum amplitude.

3. The resistance offered by water to the rolling motions of ships is most influential in limiting rolling, and any means available for increasing that resistance and the "rate of extinction" of rolling must be beneficial.

4. For any ship the condition tending to produce the heaviest rolling is that when the waves pass her at a rate which synchronizes with her natural period of oscillation in still water. Apart from the action of resistance, the successive impulse of a regular series of synchronizing waves would increase the amplitude of the oscillations so readily that any ship must be capsized in a very short time.

5. Every ship, no matter what her size and period of oscillation, is liable to this condition of synchronism. It may occur with a beam sea of period identical with her own, or it may

result from the course and speed being such as to produce synchronism with waves of dissimilar period. In fact, allowing for the infinitely varied conditions of sea necessarily encountered, no ship can altogether escape from heavy rolling.

6. Experience has confirmed the deduction from mathematical investigation that ships which are slowest in their still-water oscillations, as a rule, are steadiest at sea.

7. For slow-moving ships, of long still-water periods, the condition of synchronism is sometimes reached in a long and very low swell, with practically no wind. This swell results from waves which have traveled long distances, from the storm region in which they were formed, and in transit have been greatly degraded in height, while retaining considerable length and period.

8. When rolling is set up in ships by the passage of waves, it may be anticipated that the heavier ships, with greater moments of inertia, will accumulate the largest angles of oscillation.

This is really understood, apart from mathematical demonstration. Each wave as it passes tends to increase the oscillation, and the increment is dependent upon the wave slope, while it is little influenced by the size or weight of the ship (usually her breadth) presented to the wave advance. On the other hand, the effect of water resistance in limiting oscillation depends, as previously explained, upon the relation of its moment about the axis of rotation to the righting moment and moment of inertia of the ship. That is to say, the moment of the water resistance becomes relatively less as the weights and inertia of ships increase, and consequently in two ships of identical period exposed to the same waves (say, synchronizing in period, or nearly so) the larger ship of greater inertia will reach larger angles of oscillation.

Bearing in mind these conclusions from long experience, we may pass to the summary of facts respecting the actual behavior at sea of the *Royal Sovereign*.

During the first eight months the ship was on service there was no report in regard to her rolling, and the only statement in regard to her behavior was to the effect that off the coast of Ire-

land, in the Atlantic swell, she had compared very favorably with vessels of the *Admiral* class.

In January, 1893, during a passage from Plymouth to Vigo, the ship encountered a long, low swell, which set her rolling, and in the cruise which followed, to Madeira and back, very similar conditions prevailed. The striking feature in this case, which naturally attracted great attention, was that so large a ship should be set rolling by what would ordinarily be treated as a smooth sea, when smaller ships of less period were practically unaffected. Careful observations were made of the character of the swell which produced these results. It was found that with lengths varying from 450 feet to 700 feet were associated heights varying from 6 feet to 12 feet, and it became obvious that these waves were so related in period to the period of the *Royal Sovereign* that rolling necessarily resulted, while ships of less period remained steady. This was no new phenomenon, as I will explain hereafter.

The actual angles of rolling on this occasion were not considerable, as the following summary of observations will show:

INCLINATIONS ON EACH SIDE OF VERTICAL.

	Mean. Degrees.	*Maximum. Degrees.*
January 18, 1893, . . .	4.6	9
January 19, 1893, . . .	8.5	13.5
January 20, 1893, . . .	2	4.5
February 11, 1893, . . .	8.5	16
February 12, 1893, . . .	7	11

It will be understood, of course, that the maximum angles of inclination only occurred at intervals, the rolling passing through "phases;" mean inclinations are the fairer measures of average behavior.

After this cruise under the conditions described, no further reports of rolling were received until the end of December, 1893. The squadron was then proceeding from Gibraltar to Arosa Bay, and encountered a heavy swell, in perfectly fine weather, with an otherwise smooth sea. The apparent period of

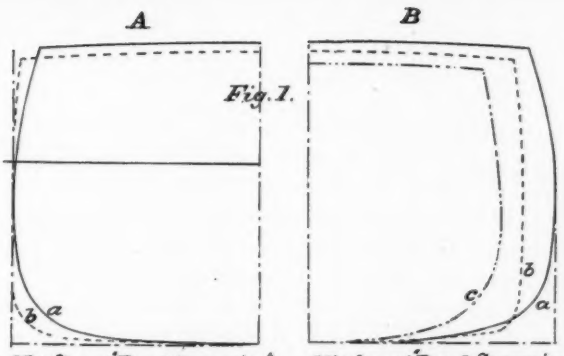


Fig. 1.

a Mid. Sec. of "Royal Sovereign".
b " " " Atlantic Liner.

a Mid. Sec. of "Royal Sovereign".
b " " " Atlantic Liner.
c " " " Sultan.

Curves of Stability for H.M.S. "Royal Sovereign".

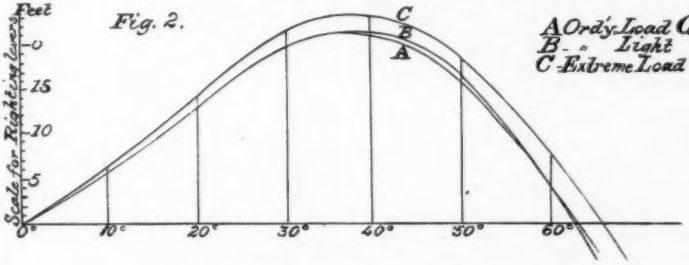


Fig. 2.

A Ordy. Load Condition
B - " Light "
C - Extreme Load "

Curves of Stability for H.M.S. "Monarch".

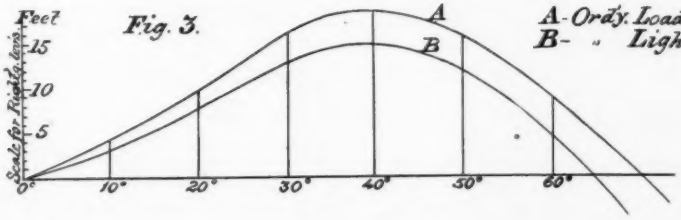


Fig. 3.

A - Ordy. Load Condition.
B - " Light " .

Curves of Stability for H.M.S. "Devastation."

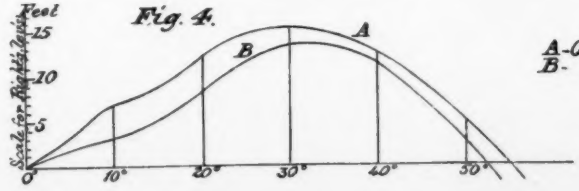


Fig. 4.

A - Ordy. Load Condition.
B - " Light "

the waves was found to vary from 13 seconds to 15 seconds; the period of oscillation of the *Royal Sovereign* class (double-swing) being 15 seconds to 16 seconds. The lengths were estimated at 650 feet to 960 feet, and the heights at 10 feet to 15 feet. In this case, therefore, there was a close approach to synchronism in the periods of the waves and the big ships, the *Empress of India* being in company; while the quicker-moving ships were more advantageously situated. In the following table are given a set of simultaneous observations for the ships of the squadron:

INCLINATIONS ON EACH SIDE OF THE VERTICAL.

	Mean. *Degrees.*	*Maximum.* *Degrees.*
Royal Sovereign,	7	11.5
Empress of India,	5.9	10.5
Immortalité,	2.8	5
Narcissus,	3	7
Bellona,	6.2	13.2

The *Rodney* was in company, but did not take simultaneous observations on the occasion described in the table.

Taking all the observations made at various times during this day, and disregarding changes which may have occurred in the course and speeds of the ships, or changes in the state of the sea, the following summary represents the facts:

MEAN INCLINATION ON EACH SIDE OF THE VERTICAL.

	Degrees.
Royal Sovereign,	6.8
Empress of India,	9.4
Immortalité,	3.
Narcissus,	3.3
Rodney,	5.8
Bellona,	6.1

In order to illustrate the general character of the motion of the large ships, the gradual increase in amplitude, attainment of a

maximum inclination and subsequent gradual decrease, Fig. 11 has been prepared from a series of actual observations.

One feature in the accumulation of rolling under the trying conditions of synchronism is the apparent suddenness with which comparatively large angles of inclination may be accumulated. Here, again, theoretical investigation anticipates observation. Apart from the action of resistance, the passage of each synchronizing wave should add an angle equal to about three times its maximum slope to the range of oscillation. In the long, low swell which synchronized with the *Royal Sovereign* class, the maximum slope was about 3 degrees only, corresponding to an increment in range, apart from resistance, of more than 9 degrees for each wave. There would be four waves passing in a minute, and it is easy to understand, therefore, in how short a time considerable angles of rolling may be reached, especially in a slow-moving ship, where resistance acquires no great moment until there is a good swing.

When the *Royal Sovereign* and *Empress of India* were placed beam on to the swell, and a series of nearly regular waves of sychronizing periods passed the ships, much larger inclinations were reached at times than are shown in the foregoing tables. The maximum inclinations recorded under these very trying conditions were about 25 degrees to 30 degrees, which is undoubtedly heavy rolling, and has naturally been made the subject of adverse remarks.

It may be well to explain, therefore, that this kind of behavior has been experienced long ago, under very similar circumstances, in vessels having unsurpassed reputations for average steadiness, and about the same periods of oscillation as the *Royal Sovereign* class. In this connection I will venture to quote a passage from my "Naval Architecture," published nearly twenty years ago:

"No feature in the behavior of ships is better established than that heavy rolling results from equality, or approximate equality, of the period of the ship and the period of waves, even when the waves are very long in proportion to their height. Many facts might be cited in support of this statement, but a few must suffice. . . . . Vessels of the *Prince Consort* class were made

to roll very heavily by an almost imperceptible swell, the period of which just synchronized with that of the ships [for a double roll]. On one occasion the *Achilles*, a vessel having a great reputation for steadiness, rolled more heavily off Portland in a dead calm than she did off the coast of Ireland in very heavy weather. Mr. Froude reports a very similar circumstance as having occurred [in the channel] during trials with the *Active*. And, lastly, during the cruise of the combined squadrons in 1871, when the *Monarch* far surpassed most of the ships present in steadiness in heavy weather, there was one occasion when, doubtless through the action of approximately synchronizing periods, she rolled more heavily in a long swell than did the notoriously heavy rollers of the *Prince Consort* class."

The *Hercules*, as a rigged ship, was remarkable for average steadiness of platform. Yet on some occasions, in a swell very similar to that which set the *Royal Sovereign* class in motion, she rolled to angles of 25 and 30 degrees on each side of the vertical. Similar experience has been obtained with the *Inconstant* and other vessels.

These instances are not quoted for the purpose of diminishing in the least degree the important influence upon fighting efficiency of steadiness of platform in war ships. It is not a possibility, however, to secure absolute steadiness under all conditions of sea, for the reasons given above. The designer, therefore, must fall back on experience, and endeavor to secure an average steadiness. His aim is to give to a new design about the same period of oscillation as was possessed by ships of proved good behavior. In the design of the *Royal Sovereign* class, as already explained, it was intended to approximate to the period of ships like the *Hercules* and *Monarch*. This has been accomplished, and it is important to note that in the ships as they now go to sea, by additions of coal and feed water, the stiffness has been increased, and period correspondingly quickened. With the weights on board originally contemplated as "normal," the period would be about 7 per cent. greater than in the deep-laden ships. As they now have periods of about 15 seconds when deeply

laden, this would mean an increase of about 1 second in the time for a double swing.

This may be thought a refinement by some persons, but it is by no means the case that such differences in period do not affect behavior. In fact, observations made in the *Royal Sovereign* class, with varying conditions of stowage of coal and feed water, show such changes in stiffness and period to have quite appreciable effects.

Small changes in course and speed relatively to synchronizing waves also produced marked results on rolling. As an illustration, I may take a recent observation made by the *Royal Sovereign*. With swell abeam, the mean inclination to the vertical was about 9, and maximum (occasionally reached) 18 degrees. With the swell two points before the beam, the mean inclination was about $5\frac{1}{2}$, and the maximum $10\frac{1}{2}$ degrees. With swell four points before the beam, the mean inclination was about 4, and maximum $9\frac{1}{2}$ degrees.

Further reports on observations of rolling were made after the passage of the Channel squadron from Vigo to Madeira in January last. There is no new feature requiring detailed notice. Throughout, there was a long, low swell, or a confused sea. On the whole, the *Royal Sovereign* class rolled less than in the December cruise, the wave periods not approaching so closely to synchronism. An analysis of the figures shows that the mean inclinations to the vertical averaged about 6 degrees for the class. Inclinations exceeding 15 degrees are not reported for the *Empress of India* during the period of observations, and are only twice mentioned for the *Royal Sovereign* and three times for the *Resolution*. On one occasion the *Resolution* reached an inclination of about 31 degrees, and the *Royal Sovereign* on one occasion reached 21 degrees. These were altogether exceptional rolls, and must have resulted from some special condition of the sea at the time, although no details appear in the reports.

All the reports on the class speak of the rolling motion of the ships as remarkably easy, free from jerks, and with quick recovery from extreme inclinations. This was anticipated from their form, period of oscillation and conditions of stability.

Summing up the facts, so far as they are in my possession, it appears that during the period of her service, now approaching two years, the *Royal Sovereign* has, on the whole, proved herself steady and well behaved at sea. The heaviest rolling of the class has occurred under the conditions of a long, low swell synchronizing, or approximately synchronizing, with their period of oscillation. In this respect the class resembles the *Hercules*, *Monarch*, and other ships of remarkable average steadiness and about equal period. Under such circumstances, the great inertia of the *Royal Sovereign* class, inevitable with their offensive and defensive arrangements, occasionally tends to produce considerable angles of inclination. A very moderate change, of course, produces considerable reduction in rolling.

Unsteadiness of platform under these conditions is clearly a drawback. There is no question of danger arising, however, as is explained hereafter. The edge of upper decks has been only just awash when the rolling has been heaviest in a long swell. Experiments are about to be made with bilge keels in one of the vessels, in order to determine whether the increase in resistance thus obtained will sensibly diminish rolling.

All past experience shows that ships having periods of 15 seconds to 16 seconds for a double swing are, as a rule, exceptionally steady at sea. The *Royal Sovereign* class have about that period, and there is nothing whatever in their under-water form or stability to make their behavior differ from that of their equally slow-moving predecessors. Their greater inertia is the new and inevitable condition of the designs. Further experience is required to determine the relative steadiness of the class under average conditions of sea.

In concluding this section, I desire to place on record our great indebtedness to the naval officers serving in the *Royal Sovereign* class for the valuable observations they have made and recorded. Captain Hammill, of the *Royal Sovereign*, deserves special thanks for the information he has afforded, as well as for many valuable suggestions. Having to go beyond precedent necessarily involves new conditions. Whatever care or skill may have been bestowed on a design, nothing but actual experience

can show whether or not it is successful. Much as we have learnt respecting the behavior of ships, there are still many obscure points, and in elucidating these naval architects must depend upon naval officers for facts as to the conditions of sea and the performances of ships.

7. *Behavior of the Resolution in December, 1893.*—The case of the *Resolution* is quite distinct from that of the other ships of the class, and it has been so prominently before the public that I propose to state the facts briefly. It will be understood that a full discussion of the incident is not intended, nor indeed is it possible for one holding my official position to attempt it, for reasons that will be obvious.

The *Resolution*, a newly commissioned ship, left Plymouth at 2 P. M. on December 18 last to join the Channel Squadron at Gibraltar. On December 23, at 9 A. M., she arrived at Queenstown.

According to the newspaper accounts, the ship had been seriously strained in structure by heavy rolling, reaching to angles of 30 or 40 degrees from the vertical. Dangerous leaks had been developed, and she could only be kept afloat by continued pumping. Considerable repairs were said to be required, and it was alleged that she was in such a state that she could hardly proceed without risk as far as Plymouth or Portsmouth before being taken in hand. Graphic accounts followed of the terrible weather which had been encountered, of the unsatisfactory behavior of the ship, the great discomforts suffered by all on board, and the immense quantities of water which were said to have passed down into the engine rooms and stokeholds. In short, the impression was produced, and possibly still remains in many minds, that the *Resolution* had a narrow escape from disaster, and her behavior was considered the more unsatisfactory seeing that the torpedo gunboat *Gleaner*, which was in her company, had proceeded to her destination, while the big ship put back.

Now for the facts. It has been stated by the representative of the Admiralty in the House of Commons that the captain of the *Resolution*, in the exercise of his undoubted discretion, put back to Queenstown because he considered it the wiser course, having

regard to the possible continuance of bad weather, and the quantity of coal remaining on board.

The ship proceeded to Devonport, and a careful survey was made by the officers of Devonport Dockyard, who had nothing whatever to do with the construction or fitting out of the *Resolution*. This survey established the entire absence of any working or weakness in her structure. Various fittings on or above the upper deck and outside the ship were damaged or washed away by the sea. The bridges, which are merely light superstructures at a considerable height above the shelter deck, showed some small signs of movement, but nothing of any importance. There were no serious leaks anywhere. One or two places were found where small fittings had been attached to the sides, and the rivets were not absolutely water-tight. A few local defects in deck planks had permitted trivial leaks to occur here and there. Some small leakage also occurred at gun ports, deck pipes, and other openings provided with covers that had been thoroughly tested during construction, but were apparently not so well secured as they might have been when the storm was encountered. In dealing with such fittings, experience counts for much, and a newly commissioned ship is always at a disadvantage. Other fittings which were in place need not have been there under the circumstances, arrangements having been made for stowing them in safe positions at sea, or when bad weather was anticipated. The covers to the forward barbette gun-wells were washed away by the sea, not because they were not strong enough, but because they were not properly secured; and this circumstance led to the entry of water through the barbette.

The best possible evidence to those conversant in such matters of the trivial and unimportant character of the defects arising from this experience of the *Resolution* is found in the fact that they were all made good—including repairs to damaged fittings and renewals of fittings lost—for the sum of 440*l*.

There is no evidence that large quantities of water, much less dangerous quantities, passed into the interior. Before the hatches were battened down amidships a sea broke on board, and some water found its way down to the main deck and into the engine

room, but this did not recur after the battening down. Discomfort there was, no doubt, during the three days the ship was battling with the storm; but that is a common experience, and not peculiar to warships. It has come to my knowledge that two large passenger steamers which were crossing the Bay about the same time had a very similar experience, but they continued their voyages, and I have not seen any comments in the press in regard to their behavior. The storm was undoubtedly severe, and the sea was exceptionally high and steep for a considerable time.

After passing Ushant early on the morning of the 19th, the *Resolution* had a swell on the beam, which caused her to roll considerably. The wind, which was south at starting, veered during that morning to west, and then to northwest. She was kept on her course for seven or eight hours, until about 7 P. M. Two exceptionally heavy rolls then occurred, and the sea broke over the upper deck amidships. The hatches were not battened down at the time, and water passed through them on to the main deck and into the engine room, as above described. After this happened the ship was brought head to sea, gear secured, and battening down completed. She then resumed her course; but, as she again began to roll considerably, she was once more brought head to sea and there kept steaming slowly. This happened about 9 P. M. on December 19.

The swell increased and the wind freshened during the night. By 8 A. M. on December 20 it was a whole gale, with tremendous squalls. Measurements of the waves were made that day with all the care possible under the circumstances. Accurate measurement, especially of length, was hardly possible. The wave heights were obtained by horizon observations, and are probably more nearly correct. Heights of 42 feet were observed from hollow to crest, and lengths of 300 feet from crest to crest. The exceptional steepness of these waves will appear from the fact that extensive observations have fixed the average height of large Atlantic storm waves at one-twentieth of the length, so that waves 42 feet high would be over 800 feet long. Similar observations give 15 feet as the average height for waves 300 feet long.

No doubt in this instance, as in many others, where the wind has veered during a long-continued storm, there were independent series of waves running in different directions and superposed on one another, which would account for the height and steepness of the waves observed.

The ship was kept head to sea until 4 P. M. on December 21, when the swell had begun to drop somewhat, but was still from the northwest. Measurements of waves that morning showed heights of 26 feet to 30 feet, and lengths estimated at 280 feet—still a very steep and heavy swell. A northerly course was then shaped; and, as the swell decreased gradually during the night, on the morning of the 22d the ship proceeded to Queenstown. On arrival she had over 450 tons of coal on board, having left Plymouth with about 790 tons.

There are two stages to be considered in this narrative. First, that during which the ship was exposed for some hours to a swell abeam, described as moderate, but said to produce occasional heavy rolling. This resembles the case above discussed for the *Royal Sovereign* class; and in his report the captain of the *Resolution* expressed the opinion that the cause of rolling was approximate synchronism between the period of the ship and that of the waves.

The second stage is that where the ship was kept head to the sea. Her behavior under these circumstances is reported to have been most satisfactory. She was very buoyant, rode well over the very heavy sea, and pitched easily. At times she rolled considerably, which is not remarkable when the state of the sea and its confused character are taken into account.

It is reported also that under these conditions the oscillations were quite different from those which occurred when the swell was abeam. In the latter case the rolling gradually increased, reached a maximum, and then gradually diminished—completing a "phase," in fact, like that represented in Fig. 11. Head to sea, the ship is stated to have "lurched" at once, or nearly so, to her maximum inclination, and then to have gradually lessened her swing in succeeding oscillations. This is what would be anticipated from the position she occupied in relation to the

short, steep sea, and the pitching and 'scending motions impressed upon her. Under these circumstances there would necessarily be great variations in the distribution of the buoyancy as compared with still water, and considerable temporary reductions in transverse stability, which would account for the occasional heavy lurching. Moreover, there must have been very severe longitudinal bending moments on the structure, and it is most satisfactory to find that there was no indication whatever of working or weakness.

Another fact of great importance must be stated. There is no trustworthy evidence of the angles through which the *Resolution* rolled on this occasion. The only observations recorded were made with a short, quick-moving pendulum placed in the chart house, at a height of 35 feet to 40 feet above water. I need not dwell upon the possible errors of pendulums so placed in a ship rolling among waves. Cases are on record where the indications of pendulums, similarly placed in ships of about the same period, have been twice the true angles of oscillation, observed simultaneously by trustworthy methods. In fact, no experienced naval officer trusts pendulum observations, spirit levels, or gravitational instruments of any kind for use at sea.

One cannot fix precisely what was the actual error of the pendulum in the *Resolution*. But it is certain that its movements placed at such a height must have grossly exaggerated the rolling of the ship. When angles of inclination of 30 degrees or 40 degrees are said to have been indicated by, or estimated from, the movements of the pendulum, the actual angles must have been much less, and may have been only half what was shown on the scale.

It is important to note also that, even had these great angles of inclination been reached, there would have been no reason for apprehension of danger to the ship. The curves of stability (Fig. 2) show that the maximum righting moment occurs at an inclination to the vertical of 37 degrees, and that instability is not reached until the inclination exceeds 60 degrees. There is a popular impression, no doubt, although an absolutely erroneous one, that if a ship is inclined past her angle of maximum sta-

bility she will capsize. To this Institution I need not explain that even in a ship with large sail power, and exposed to squalls of wind, there may be no danger in angles of inclination much exceeding that at which the righting moment reaches its maximum. But when, as in the case of the *Resolution*, there are no sails, and the inclining force of the wind on the broadside is relatively so small, there is a much greater margin of safety.

In Fig. 7 a cross-section of the *Resolution* is given for an inclination, in still water, of 40 degrees to the vertical. At this inclination the righting moment, for all practical purposes, may be considered as maintaining its maximum, and has a value of more than 30,000 foot-tons. This is the greatest angle of roll which, by pendulum or estimate, has been attributed to the *Resolution*. For the reasons just stated, it is practically certain it was never approached. Had it been approached, the upper deck must have been invaded by the sea to a much greater extent than actually happened.

Among waves, the rise of the sea, as well as the rolling of the ship, necessarily influences the amount of water which comes on to the deck. This fundamental truth is often overlooked, and in some discussions on the *Resolution's* behavior, this has been done. Simply to illustrate the well-known fact, Figs. 8 and 9 have been prepared. The latter shows the *Resolution* beam-on to trochoidal waves of the heights and lengths said to have been observed on December 20, but to which she was then kept bow-on. The maximum slope of the waves is 26 degrees, and an inclination to the vertical of 8 degrees would then immerse the top of the bulwarks. In this position the corresponding angle on the curve of stability for measuring the righting moment would, of course, be 34 degrees.

Fig. 8 shows the *Resolution* beam-on to an Atlantic storm wave 600 feet long and 30 feet high, the maximum slope being 9 degrees. Here an angle of inclination to the vertical of 25 degrees would bring the top of the bulwarks awash.

In practice, of course, waves have not the regular forms shown on the diagrams, nor are successive waves of identical size and height. The "rise of the sea" on occasional waves in a series

often considerably exceeds the average rise, and in a confused sea formed by superposed series these variations in height are very striking. Even with my limited opportunities for studying wave phenomena, this has been much impressed on my mind, and experienced naval officers have frequently drawn attention to the same fact when the assumptions made in the mathematical theory of rolling have been explained to them.

Fig. 10 has been constructed on the hypothesis of two superposed series of waves, one 600 feet in length by 20 feet in height, and the other 300 feet by 20 feet in height. At certain times the ship on such a compound wave, if broadside on, might immerse her bulwarks at an inclination of 18 degress. If end on, the wave crests passing rapidly, might also rise considerably above the bulwarks, and come on to the upper deck.

The presence of water in considerable quantities, at times, on the upper deck of the *Resolution* cannot, therefore, be regarded as any proof that the ship approached the angles of inclination indicated by the pendulum; much more moderate angles, under the circumstances described, would have produced this result.

It is undoubted that the ship at times rolled considerably, and that the *Gleaner*, although of small dimensions, made better weather of it when beam-on to the swell first encountered. The *Gleaner* is said to have "risen over the beam swell and sea like a cork," which is what would be anticipated from the fact that she is a quick-moving ship, her period being only one-third to one-fourth that of the *Resolution*. On the swell which synchronized with the movements of the latter and accumulated rolling, the *Gleaner* could move so quickly as practically to accompany the waves. Every one knows that under such circumstances the smaller ship is likely to have an advantage; but it is equally true that under most conditions the larger and slower-moving ship will be better behaved than the smaller, and more capable of maintaining her speed.

The *Resolution* was, in fact, put to a severe test at the very commencement of her service, when those on board had not grown familiar with the vessel, and particularly with the fittings provided for use in rough weather. She was not fully prepared

when the sea first broke on board, and the washing away of certain fittings which were not well secured, or which were left in place instead of being stowed, permitted water to pass below. This involved discomfort, but not danger; and the statements to the contrary are unfounded. So are the accounts that were published in regard to the straining and leaks produced by the rolling. As to the extent of the rolling, there is no exact information, but there was undoubted exaggeration in the published report.

8. *Performances under Steam.*—In the design it was proposed to develop 9,000 horse power on the natural-draft contractor's trial of eight hours, the corresponding speed being about 16 knots; and on the four hours' forced-draft trial to develop 13,000 horse power, the corresponding speed being 17½ knots.

The *Royal Sovereign*, being the first completed ship of the class, was chosen for the purpose of comparing estimates with actual performances. She was brought to her designed load line, and was practically ready for service when the trials were made. On the eight hours' trial the mean indicated horse power was 9,667, with .39 inch air pressure in stokeholds; the revolutions averaged 97 per minute. During the trial four runs were made on the measured mile at Stokes Bay. The mean number of revolutions per minute on these runs was 99, the mean power 9,780 horse power, and the mean speed 16.77 knots. On this basis, the mean speed for the eight hours was 16.43 knots, including the turns made as the ship ran her trial.

It may be worth mentioning that the speed by log on this trial was 16.32 knots, showing a fair agreement with actual performance. Of course, this agreement was accidental, since it is fully recognized that many circumstances seriously affect the indications of the log. But it is the custom of the service to have the log running when ships are under weigh, and to record their indications. Not unfrequently in published accounts of trials one sees the log speeds given without any notation of the mode of speed measurement; and in some instances I have observed detailed criticisms of relative performances based on log speed records which were altogether untrustworthy. A far better

method of approximating to speeds when making long runs is, undoubtedly, to determine the curves of revolutions for speeds by progressive trials, and to use these results, instead of any log yet available.

On the forced-draft trial, the *Royal Sovereign* was run up Channel and back in deep water, and over a known distance. For three hours' continuous steaming the mean revolutions were 106.3, the mean development was 13,360 horse power, with 1.6-inch air pressure in stokeholds, and the speed (by revolutions) was 18 knots. I may add that this result was checked by a comparison of the "coefficients of propulsion" obtained from the model experiments, at 16¾ and 18 knots.

It will be seen, therefore, that the anticipations of the design were fully realized in regard to the speeds corresponding to the specified powers, and that the actual development was above the specified both for natural and forced draft.

As all the ships are of identical form, and the screws of the same type and dimensions, nothing was to be gained by repeating the trials for speed, and incurring the expense of ballasting in order to bring them down to their designed load lines. It was decided, therefore, that the contractors' trials on the remaining ships should be limited to the development of power under the specified conditions, care being taken to immerse the screws sufficiently. The ships were run at draughts and displacements lighter than their completed draughts, varying from 24 feet 1 inch mean draught and 12,100 tons displacement up to 26 feet 11 inches mean draught and 13,500 tons. The *Royal Sovereign* was tried at 27 feet 6 inches draught and about 14,200 tons displacement.

On the natural-draft trial of eight hours' continuous steaming, the developments varied from 9,180 to 9,600 horse power, the air pressures in stokeholds varying from .2 inch to .5 inch of water. The specification provided for 9,000 horse power with .5-inch air pressure.

In the later ships it was decided not to push the forced-draft trials so far as had been done in the *Royal Sovereign*, but to limit the development to 11,000 horse power. With this limitation it

was possible to set the valves so as to favor economy in the use of steam at ordinary working speeds, and to increase the efficiency under general-service conditions, when the air pressure in the stokeholds would be not more than .5 inch. With ordinary coal and stoking the specified natural-draft power (9,000 horse power) would not be exceeded and not often reached, so that 11,000 horse power gave a good margin in the engine department. The steam-producing power of the boilers, of course, remained unchanged.

The mean development on the four hours' trials varied from 11,300 to 11,600 horse power. The air pressure varied from .35 to 1 inch.

No trustworthy observations of speed were made, but the logs were run as usual. Attempts have been made by some persons to analyze and compare the results, on the assumption that the speeds were correctly measured. All these attempts simply represent so much waste work.

In my paper of 1889 I explained that the greater length of these ships would favor economy of power at the higher speeds, as compared with what had been accomplished in vessels previously built. This has been fully realized on trial, and the following figures are of interest:

The *Trafalgar*, 345 feet in length, and of 12,000 tons displacement at the trial draught (27½ feet mean), required 8,440 horse power for 16.22 knots, and 12,900 horse power for 17.28 knots. The *Howe*, 325 feet in length, 9,600 tons displacement at the mean draught of 26 feet 8 inches, required 8,230 horse power for 15.9 knots, and 11,600 horse power for 16.9 knots. The *Royal Sovereign*, 380 feet in length, and about 14,200 tons displacement on the same mean draught as the *Trafalgar*, required 9,780 horse power for 16.77 knots, and 13,360 horse power for 18 knots.

Much might be said in further illustration of the point, which was discussed at length five years ago, but these figures must suffice at present.

From personal observations made during the trials of these

ships, I can testify that at the higher speeds, even in smooth water, the greater freeboard of the *Royal Sovereign* class is a source of sensible economy in propulsion. Against the high bow the waves can maintain themselves when once formed, whereas in the vessels of low freeboard a wall of water is gradually raised above the upper deck level, until it reaches such a height that it falls over upon the deck, and has to be recreated. In a moderate sea the conditions are still more favorable to the ships with high bows, and the maintenance of speed in rough water is greatly superior.

We are accustomed to adverse criticism of measured mile performances, which are of great value for many purposes, although no one supposes them to represent what can be done on long-distance steaming. Fortunately it is now possible to refer to a thorough test of the *Royal Sovereign* on a trial made from Plymouth to Gibraltar, in October last. The run extended over 72 hours, and was made in fair weather, except for a few hours towards the end, when head winds and some sea were encountered. The mean speed for the 72 hours was 15 knots; the mean power was 8,180—90 per cent. of the specified power. This is an excellent result, most creditable to the engineer officers and their staff. As regards the speed, it is proper to note that the ship started with a considerable load of coal and stores, drawing about 28½ feet, and having a displacement of about 14,900 tons. Her consumption of coal for steaming was 484 tons, and for all purposes about 500 tons. Hence her mean displacement for the run was about 14,650 tons. At this speed, therefore, the maximum coal supply (1,450 tons) would suffice to cross the Atlantic at a speed of 15 knots. This is probably an unrivalled performance for a battleship. The rate of coal consumption was 1.84 pounds per indicated horse power per hour.

It is interesting to compare this performance on a long run at sea with that of some other battleships when tried on the measured mile for short periods and with everything at its best, as an indication of what has been accomplished in the direction of economical propulsion.

Curves of Righting Moments in Ordinary Load Conditions.

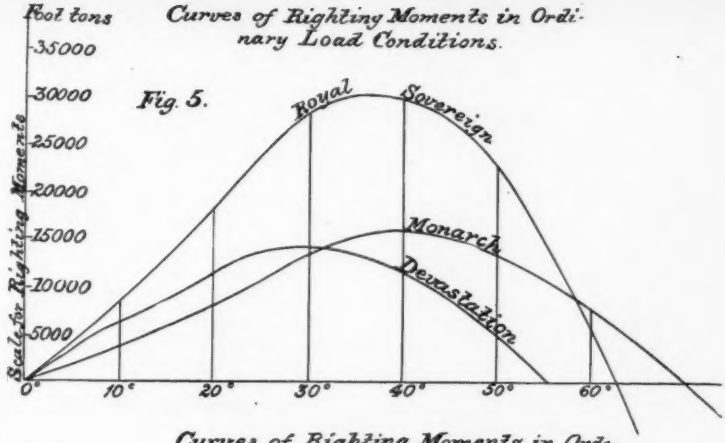


Fig. 5.

Curves of Righting Moments in Ordinary Light Conditions.

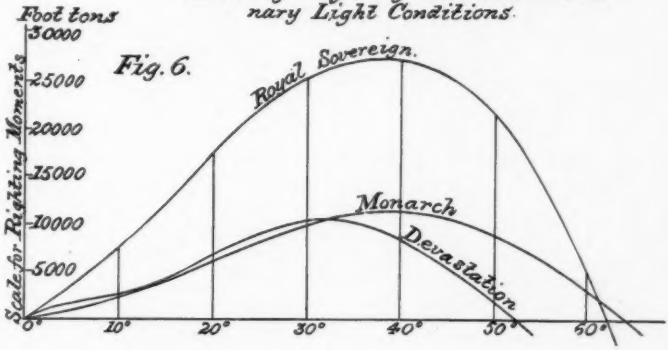


Fig. 6.

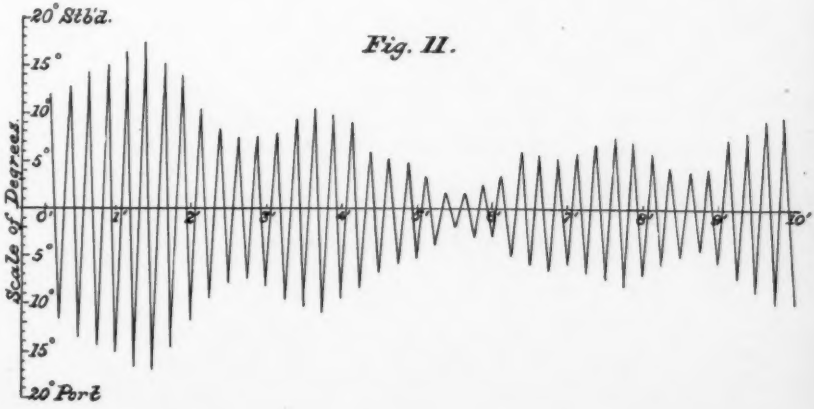
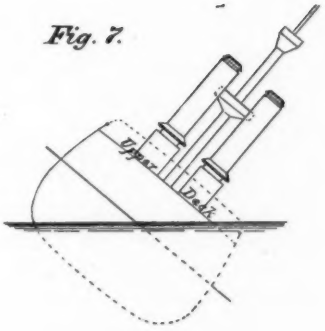
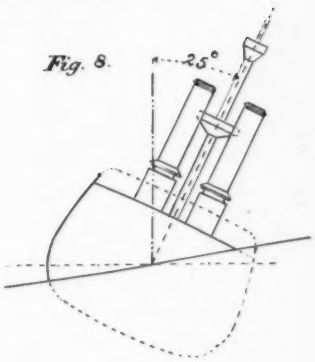
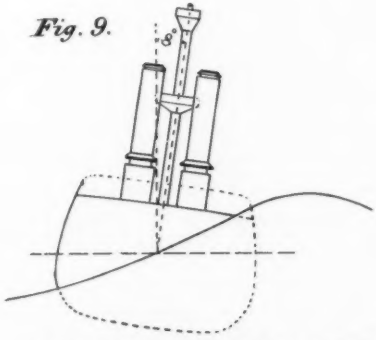


Fig. 11.

H. M. S. RESOLUTION.







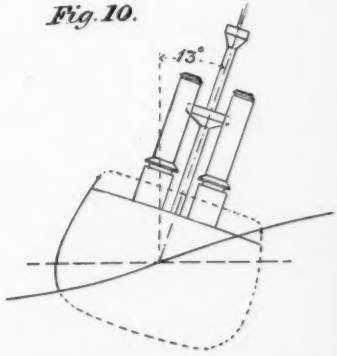


Fig. 7. If inclined at 40° in smooth water.

Fig. 8. Broadside on to a wave 600 feet long, showing inclination necessary to put her bulwarks under water.

Fig. 9. Broadside on to a wave 300 feet long, 42 feet high, showing inclination necessary to put her bulwarks under water.

Fig. 10. Broadside on to a wave 300 feet long, 20 feet high, superposed an another wave 600 feet long and 20 feet high.

	Length.	Mean Draught.	Displacement.	Indicated Horse-power.	Speed in Knots.	
	ft.	*ft. in.*	*tons.*			
Royal Sovereign	380	28 2	14,650	8,180	15	Run to Gibraltar.
Alexandra	325	26 1½	9,430	8,615	15	Measured mile trials.
Dreadnought	325	26 10	11,000	7,927	14.24	
Hercules	325	24 8	8,676	8,529	14.69	

At cruising speeds also the *Royal Sovereign* has done well. When steaming at a speed of 7.6 knots per hour the expenditure of one ton of coal suffices to drive her enormous weight (over 14,000 tons) more than 7 knots, less than 1,000 horse power being indicated. For 10 knots speed about 2,500 horse power is required, and an expenditure of one ton of coal would drive the ship about 4½ knots in fair weather with bottom clean.

Trials have been made as to the most economical mode of propulsion at very low speeds, with the result that it has been found preferable, even at speeds of 5½ to 6½ knots per hour, to keep both engines at work rather than to work one engine only and use the helm to keep a course.

This is a suggestive experiment, taken in connection with the economy anticipated in some quarters by adopting triple screws, in order to work only the central engine at low speeds.

Similar trials made in other twin-screw ships have given confirmatory results.

9. *Manœuvring Powers.*—The vessels of the *Royal Sovereign* class have proved very handy for their dimensions, comparing favorably with many ships of less size and weight. They are perfectly under control, answering all movements of the helm even at low speeds. One remarkable proof of their handiness is to be found in the fact that vessels of the class have proceeded under their own steam alone, without the assistance of tugs, from Plymouth Sound into the Hamoaze, a passage involving very awkward turns in limited space and a strong tideway. Turning trials have been made with several of the ships, some

fully laden and others light. It need hardly be said that differences of draught and trim affect performance.

Taking the sea-going condition, it appears that the tactical diameter is about five times the length, or about 1,500 feet. This tactical diameter is practically constant for all speeds. The time of turning varies, therefore, nearly inversely as the speed.

At 12 knots the course can be reversed in about 3.4 minutes; at higher speeds in proportionately less time.

The actual space required for turning is practically the same as that required by a number of battleships from 40 feet to 50 feet shorter, and 2,000 to 3,500 tons less displacement, and taking the ratio of tactical diameter to length, the big ships compare favorably with any battleship of recent construction and high speed; notwithstanding their great inertia due to the weights and positions of the barbettes and heavy guns.

Relative Size and Cost of Royal Sovereign Class.—In my paper of 1889 I explained that the comparatively large dimensions and displacements of the *Royal Sovereign* class, resulted from the fact that they were designed to carry great weights of armor, armament, coals and equipment, at high speeds. As compared with first-class battleships recently built, or now building, for foreign navies, they are of large displacement tonnage, and somewhat greater draught. As these foreign ships are intended to have about the same speed, while in armor and armament they are designed to meet our ships on fairly equal terms, it has been assumed in some discussions of the subject that our ships are unnecessarily large for their intended service.

To this Institution it is unnecessary to explain how fallacious a standard of measurement is displacement tonnage, apart from a full statement of what is included therein. One requires to know not merely what numbers and weights of guns are carried, what are the thicknesses of armor and areas protected, but also what are the weights of ammunition, coal, stores and equipment included in the nominal displacement—in other words what is the total load carried at the maximum estimated speed.

Fortunately, I have been able to make such a comparison between the details of the designs of the *Royal Sovereign* class

and those of some foreign battleships of the same date of nearly equal length, and of about 12,000 tons displacement. Without going into particulars, I can state broadly that the total load in our ships reckoned in the displacement, and carried at the maximum speeds, exceeds that of the foreign designs by about 1,600 tons. This excess is almost entirely made up on the items of coal, ammunition, stores and equipment.

Everyone familiar with ship designing knows how an addition to the load, with constant speed, leads to a much greater increase than itself to the displacement. But apart from this it will be seen that if the *Royal Sovereign* carried 1,600 tons less than she does—that is, had an equal load with that carried by the foreign ship—she would be about 12,500 tons displacement, and would draw about 2½ feet less than she does. Of course at this lighter draught she would have a greater speed also.

This is only a typical case. In the Royal Navy the established policy is to give to our ships greater sea-keeping power, more coal, more ammunition, more stores. Hence, with equal skill in design, equal weights of armor, equal numbers and weights of guns, and equal speed, our ships must be of greater displacements.

Thanks to our superior economy in construction, the cost of our ships compares favorably with those of foreign ships. The dockyard-built *Royal Sovereigns* cost about £770,000, exclusive of establishment charges and armaments. The corresponding cost of foreign ships of recent construction are from £950,000 to £1,000,000.

Centurion and Barfleur.—The *Centurion* and *Barfleur*, also built under the Naval Defence Act, may be briefly mentioned before concluding. These ships are designed specially for service as flagships on distant stations, and their draught limited, so that they can pass through the Suez Canal. They are wood sheathed and coppered. Their length is 360 feet; displacement, about 10,500 tons; maximum speed, about 18½ knots. On trial they have both proved excellent steamers. Their net cost (exclusive of armaments (is about £550,000.

In them we have an illustration of the price paid in reduced

offensive and defensive power, when first cost is diminished. The *Royal Sovereign* and *Centurion* may be treated as about equal in speed, manœuvring capability, freeboard, coal endurance, and the power of maintaining speed at sea. But to effect a saving of 30 per cent. in cost, it is necessary to reduce the calibers of the four heavy guns from 13.5 inches to 10 inches, and of the secondary armament from 6 inches to 4.7 inches. The maximum thickness of armor on the belt has to be made 12 inches instead of 18 inches, and that on the barbettes 9 inches instead of 17 inches.

As the designer of both classes, I can say that the work on the *Centurion* was as carefully thought out as that on the *Royal Sovereign*. The ships have been built in the same dockyard, and under similar conditions. The comparison is, therefore, fair and complete. And the broad result is that, however well adapted the *Centurion* may be for her special service, and capable as she undoubtedly is of meeting on equal terms ships of larger size, but earlier construction, she is distinctly inferior to foreign ships of the first class now building in offence and defence. Her heaviest guns cannot perforate the armor on the hulls and heavy gun emplacements of those ships. Their heavy guns can readily perforate her defence.

Seven *Centurions* can be built for the cost of five *Royal Sovereigns*. I leave to others the discussion of the question whether such a gain in numbers is any adequate compensation for loss in individual power.

[The *Repulse* has been fitted with bilge keels, and experience at sea with her, in company with other vessels of the *Royal Sovereign* class, has been so satisfactory that all the other vessels of the class are to have them.—EDITOR.]

ALBATROSS.

By Chief Engineer Isherwood, U. S. Navy.

The *Albatross* is a twin-screw steamer, designed especially for the use of the United States Commission of Fish and Fisheries. The hull is of iron, and is not wooden sheathed, for the external immersed surface of the vessel does not foul sufficiently during the short intervals of time between its being in fresh water to have warranted the additional cost of wooden sheathing and copper.

The hull is brigantine rigged with a considerable proportion of sail power; but, as the vessel comes under the category of a coasting steamer, performing its principal service in Summer when the winds are light, the sails and their concomitants might have been advantageously omitted, for the air resistance of the spars and rigging, and the immersion of the hull due to their weights, undoubtedly decreased the speed more than the sails ever increased it, leaving the first cost of the sail power and the after cost of maintaining it, together with the cost of the coal for overcoming the resistance due to the weights and surfaces of the masts, spars, rigging, sails, etc., as wholly a loss.

The twin screws, placed just in advance of the after side of the sternpost of the vessel, were adopted as a necessity in order to leave the vessel's wake as undisturbed as possible for the trawls and dredges, and their gear, which would have been much interfered with by a single screw placed abaft the sternpost. As the twin screws furnish a sort of insurance against the inconveniences of a breakdown in the motive engines, the probability being but slight that both engines would become useless at the same time, there was still less reason for burthening the vessel with sails whose only value would be found in such a contingency.

The bunkers contain 138 tons of anthracite.

The following are the principal dimensions of the hull and machinery, to which is added the performance which can be permanently maintained with anthracite as a fuel burned by natural draft:

HULL.

Extreme length, over all, feet	234
Length from forward edge of stern to after edge of sternpost, on the load water line, feet	200
Extreme breadth on the load water line, feet and inches	27–7¼
Depth of hull amidship from top of spar-deck beams to lower edge of plating, feet and inches	18–6½
Depth from load water line to lower edge of plating, feet	11.312
Depth of keel below the lower edge of the plating, inches	7⅜
Mean load draught of water from bottom of keel, feet	11.927
Greatest immersed transverse section to load water line, square feet	275.574
Area of water section at load water line, square feet	4,061.4
Displacement to load water line, tons	1,003.47
Displacement per inch of draught at load water line, tons	9.689
Ratio of the length of the hull on the load water line, from forward edge of stem to after edge of sternpost, to its breadth	7.245
Ratio of the greatest immersed transverse section to its circumscribing parallelogram	0.882
Ratio of the water section to its circumscribing parallelogram	0.736
Ratio of the displacement to its circumscribing parallelogram	0.561
Ratio of the displacement to a prism having for its base the greatest immersed transverse section, and for its height the length of the hull on the load water line from the forward edge of the stem to the after edge of the sternpost	0.636
Immersed exterior, or wetted surface of hull, to load water line, exclusive of the surfaces of the two bilge keels, keel and rudder, square feet	6,812.0
Surfaces of the two bilge keels, square feet	214.6
Surface of the keel, square feet	283.4
Surface of the rudder, square feet	41.0
Aggregate wetted surface, square feet	7,351.0
Aggregate surface in the five principal sails, square feet	4,885.0
Square feet of sail surface per square foot of the greatest immersed transverse section	17.727
Square feet of sail surface per square foot of water section	1.203
Square feet of sail surface per ton of displacement	4.868

Including the gaff topsail, flying jib, fore trysail and fore staysail, the vessel has an aggregate of 7,521 square feet of sail surface.

The bilge keels, one on each side of the keel, are 80 feet in length and 16 inches in breadth.

The weight of the hull and fittings is 412 tons.

The houses erected upon the spar deck are nearly $7\frac{1}{2}$ feet in height, occupying at their greatest breadth the entire breadth of that deck, making the total height above the water for air resistance 14 feet.

ENGINES.

There are two compound condensing engines, duplicates in every particular, and driving twin screws. The engines are direct acting and nearly vertical, being inclined $14\frac{1}{2}$ degrees from the perpendicular. Both engines have one surface condenser in common, which is placed immediately above the keelson, the center of the condenser being in the longitudinal vertical central plane of the vessel. The corresponding cylinders of both engines are placed back to back and bolted together, and the four cylinders are bolted to the condenser which thus becomes their frame or support on their inner side while their outer side is supported by inclined wrought-iron columns. The cylinders are not steam jacketed. The guides for the main crossheads are bolted to the upper part of the sides of the condenser. Each engine is complete in itself, and distinct with the exception that the condenser is in common.

The air pump (one for each engine) is single acting and horizontal; it is worked from a small special crank on the forward or free end of the crankshaft. It is a plunger pump, the hollow cylindrical plunger being open at the outboard end after the manner of a half-trunk, and the rod connecting the plunger with the crank pin is secured to the bottom of the plunger and vibrates within the half-trunk, the plunger having no other guide than a deep stuffing box.

The condenser tubes are of brass, horizontal, and are arranged in three compartments, one vertically over the other; the refrigerating water is inside the tubes and the exhaust steam surrounds them. The refrigerating water enters the lower compartment of tubes first, and passes next into the middle compartment, from which it goes to the upper compartment, and thence outboard.

Each engine has one horizontal single-acting plunger feed pump placed opposite the air pump; the axes of the two pumps being in the same straight line. The feed-pump plunger is worked direct from the air-pump plunger.

The refrigerating or injection water is supplied to the tubes of the condenser by an independent steam pump, the steam cylinder of which is 14 inches in diameter with 14 inches stroke of piston. The pump or water cylinder is double acting, 16 inches in diameter with 14 inches stroke of piston. This pump can be worked to 200 single strokes per minute; its ordinary speed is 75 strokes per minute.

The cylinder valves are operated by the usual Stephenson link and eccentrics. A reversing steam cylinder is attached. The main valves of the small cylinders are locomotive slides over which gridiron cut-off valves are placed. The main valves of the large cylinders are double ported slides; these cylinders have no separate cut-off valves, the cutting off in them being effected by steam lap.

The following are the principal dimensions for each engine:

Number of cylinders to each engine	2.
Diameter of the small cylinder, inches	18.
Diameter of the piston rod of the small cylinder, inches	3.
Net area of the piston of the small cylinder, square inches	250.935
Stroke of the piston of the small cylinder, inches	30.
Space displacement of the piston of the small cylinder per stroke, cubic feet	4.351
Space in clearance and steam passage at one end of the small cylinder, cubic feet	0.340
Clearance of the piston of the small cylinder, inch	0.5
Fraction which the space in the clearance and steam passage at one end of the small cylinder is of the space displacement of its piston per stroke	0.078
Number of steam ports at one end of the small cylinder	1.
Length of the steam port of the small cylinder, inches	13.50
Breadth of the steam port of the small cylinder, inches	1.75
Area of the steam port of the small cylinder, square inches	23.625
Length of the exhaust port of the small cylinder, inches	13.50
Breadth of the exhaust port of the small cylinder, inches	3.50
Area of the exhaust port of the small cylinder, square inches	47.25
Number of ports in the cut-off valve of the small cylinder	3.
Length of each port in the cut-off valve of the small cylinder, inches	13.50

Breadth of each port in the cut-off valve of the small cylinder, inch	0.875
Aggregate area of the ports in the cut-off valve of the small cylinder, square inches	35.4375
Diameter of the large cylinder, inches	34.
Diameter of the piston rod of the large cylinder, inches	3.5
Net area of the piston of the large cylinder, square inches	903.110
Stroke of the piston of the large cylinder, inches	30.
Space displacement of the piston of the large cylinder per stroke, cubic feet	15.621
Space in clearance and steam passage at one end of the large cylinder, cubic feet	1.0975
Clearance of the piston of the large cylinder, inch	0.5
Fraction which the space in the clearance and steam passage at one end of the large cylinder is of the space displacement of its piston per stroke	0.070
Number of steam ports at one end of large cylinder	2.
Length of each steam port of the large cylinder, inches	20.
Breadth of each steam port of the large cylinder, inches	1.5
Aggregate area of the steam ports of the large cylinder, square inches	60.
Face of pistons of all cylinders, inches	6.
Depth of packing ring in piston of small cylinder, inches	4.50
Depth of packing ring in piston of large cylinder, inches	3.75
Thickness of metal of all cylinders, inch	1.
Ratio of the net area of the piston of the large cylinder to the net area of the piston of the small cylinder	3.599
Ratio of the space displacement of the piston per stroke of the large cylinder plus the space in the clearance and steam passage at one end of the large cylinder, to the space displacement of the piston per stroke of the small cylinder plus the space in the clearance and steam passage at one end of the small cylinder	3.564
Number of tubes (seamless brass) in the surface condenser	2,394.
Outside diameter of the tubes, inch	0.625
Exposed length of the tubes, inches	66.
Aggregate condensing surface in the tubes, square feet	2,154.445
Number of air pumps to each engine (single acting)	1.
Diameter of air pump, inches	16.
Stroke of the piston or plunger of the air pump, inches	13.5
Space displacement of the piston of the air pump, per stroke, cubic feet	1.571
Number of feed pumps to each engine (single acting)	1.
Diameter of the feed pump plunger, inches	4.5
Stroke of the feed pump plunger, inches	13.5
Space displacement of the plunger of the feed pump per stroke, cubic feet	0.124
Diameter of the crosshead journal of the small cylinder, inches	3.0
Length of the crosshead journal of the small cylinder, inches	4.5
Diameter of the crosshead journal of the large cylinder, inches	3 5
Length of the crosshead journal of the large cylinder, inches	5.0

Diameter of the neck of the connecting rod of the small cylinder, inches.	3.0
Diameter of the neck of the connecting rod of the large cylinder, inches.	3.5
Length of the connecting rod of both cylinders between centers, inches..	74.0
Number of crankshaft journals to each engine.	3.
Diameter of the forward journal, inches	7.0
Length of the forward journal, inches.	8.5
Diameter of the center journal, inches..	8.5
Length of the center journal, inches..	16.0
Diameter of the after journal, inches.	8.5
Length of the after journal, inches	13.5
Diameter of the crankpin journal of the small cylinder, inches.	5.75
Length of the crankpin journal of the small cylinder, inches.	7.50
Diameter of the crankpin journal of the large cylinder, inches	7.25
Length of the crankpin journal of the large cylinder, inches	9.00
Diameter of the line shaft journals, inches	8.
Length in the vessel occupied by the two engines, feet	9.333
Breadth in the vessel occupied by the two engines, feet.	15.5
Height of the engines above axis of shafts, feet	12.5

BOILERS.

There are two return ascending flue boilers; the flues are divided into a lower group and an upper group, those of the upper group being returned above those of the lower group and the furnaces.

The shell of the boilers is in two portions, of different form. The front portion is rectangular in plan, 8 feet 6 inches in width, 7 feet 6 inches in length, and 9 feet 4 inches in height; the sides are vertical and the top is semi-circular. The back portion of the shell is cylindrical with a flat end; diameter, 8 feet 6 inches, length, 14 feet.

Each boiler contains two furnaces, and they are placed in the front portion of the shell. Each furnace is 43½ inches wide in the clear, and 6 feet 6 inches in length. The front portion of the shell has no water bottom, and the furnaces are fitted with independent cast-iron ashpans placed beneath the grates. The crown of the furnace is an arc struck from three centers with radii of 7 and 51 inches, and joining tangentially the vertical sides. The flat water space between the furnaces is 6 inches wide, including thicknesses of metal; the flat water spaces between the furnaces and the sides of the shell are 4½ inches wide, including thick-

nesses of metal; the flat water space between the furnaces and the front end of the shell is 4½ inches wide, including thicknesses of metal. The flat water space between the ashpit and the back of the front portion of the shell is 7½ inches wide, including thicknesses of metal. The bridge wall of each furnace is formed by a vertical extension of this water space; its top curves upwards, and the lowest point of the curve is 14 inches above the upper surface of the grate bars. The crown of the furnace rises from the front to the back. At the front of the furnace the vertical distance from the upper surface of the grate bars to the highest point of the crown is 25 inches. At the back of the furnace this distance is 34 inches. The upper surface of the grate bars slopes from front to back of furnace 4 inches.

The opening of the furnace door has rectangular bottom and sides, and a semicircular top; width of opening, 16 inches; height of opening, 16 inches.

Behind the bridge wall of each furnace is a combustion chamber 21 inches long in the clear, and lying wholly within the cylindrical portion of the shell. Its top is a horizontal extension of the furnace crown; its width in the clear is 43½ inches; its bottom is curved nearly concentric with the shell.

From the combustion chamber to the after smoke connection there proceed, for each furnace, five horizontal flues, one of which is 15 inches in inner diameter, another of which is 12 inches in inner diameter, and the remaining three of which are 11 inches in inner diameter. Least water space between the flues, 3 inches, including thicknesses of metal. The extreme length of all these flues is 10 feet.

The back smoke connection is separate for each furnace. It is 58 inches in extreme height; 22½ inches long in the clear, lengthwise the boiler; and 45 inches wide in the clear. The bottom and outer side are nearly concentric with the cylindrical shell. The top is horizontal and the inner side is vertical. The back smoke connections of each boiler are separated by a flat water space of 6 inches width, including thicknesses of metal, and they are separated from the back end of the boiler by a flat

water space of 4½ inches width, including thicknesses of metal. Access is had to each connection through a circular manhole of 16 inches diameter in the back end of the boiler at the bottom of the connection.

From each back smoke connection to the uptake there are returned eight horizontal flues of 9 inches inner diameter, and 16 feet 9 inches extreme length. They are distributed in two rows vertically, the four flues of the upper row being directly over the four flues of the lower row. The least water space between the flues is 3 inches, including thicknesses of metal. The vertical distance between the highest point of the crown of the furnace and the lowest point of the lower row of tubes is 6 inches, including thicknesses of metal.

The uptake is situated at the front of the boiler and immediately above the furnaces. It is in common for all the upper sixteen 9 inches diameter flues of one boiler. Its lower portion is rectangular; 8 feet wide in the clear, 25½ inches long in the clear, lengthwise the boiler, and 27 inches high. The bottom is on a level with the lowest point of the lower row of 9 inches diameter flues, and is horizontal. The lower portion of the uptake is separated from the front of the boiler by a flat water space 4½ inches wide including thicknesses of metal; and from the side of the boiler by a flat water space 3 inches wide, including thicknesses of metal. Access is had to it through a manhole at its center, 18 inches wide and 14 inches high. The upper portion of the uptake terminates at the height of 4 feet 9 inches above the bottom of the lower portion. At the top it is drawn to a semicircle of 52 inches inner diameter. The boilers are separated by a space 8 inches wide in the clear, and the diameter of the top of the upper portion of the uptake is at the center of this space.

From the top of the upper portion of the uptake rises the chimney, which, for each boiler, is a semicircle in cross section, of 52 inches diameter for a height of 14 feet above the top of the shell of the boiler. The diameter is formed by a plate of boiler iron. Above this height of 14 feet, the chimney is in common for both boilers, and has a circular cross section of 52 inches diameter.

The lower semicircular part of the chimney for each boiler is surrounded concentrically by a vertical steam drum, semicircular in cross section, 88 inches in diameter, and 14 feet in height above the top of the boiler shell. The forward point of the steam drum projects 13½ inches over the forward end of the boiler. The width of the steam drum radially is 18 inches, including thicknesses of metal.

Each boiler has its own separa s, safety valves, feed and blow valves, steam a ; and it is covered with thick felt stitched on canvas to prevent loss of heat by radiation. The crown sheets of the furnaces and the flue sheets are of steel; all the rest of the boilers is of wrought iron. The flues are all drawn seamless, the flue sheets being flanged to receive them. Each boiler is set in cast-iron chairs.

The following are the dimensions and proportions of the boilers, namely:

Number of boilers	2.
Extreme length, exclusive of steam drum, feet and inches	21–6
Extreme breadth of each boiler, feet and inches	8–6
Extreme height of each boiler, exclusive of steam drum, feet and inches	9–4
Extreme height of each boiler, inclusive of steam drum, feet and inches	23–4
Aggregate number of furnaces, in both boilers	4.
Breadth of each furnace, inches	43½
Length of each furnace, feet and inches	6–6
Aggregate area of grate surface, square feet	94 25
Aggregate number of flues in the lower group, both boilers	20.
Length of the flues of the lower group, feet	10.
Inner diameter of the flues of the lower group, inches	4 of 15, 4 of 12, and 12 of 11
Aggregate number of flues in the upper group, both boilers	32.
Length of the flues of the upper group, feet and inches	16–9
Inner diameter of the flues of the upper group, inches	9.
Aggregate heating surface in the furnaces of both boilers, square feet	224.00
Aggregate heating surface in the combustion chambers of both boilers, square feet	116.00
Aggregate heating surface in the lower group of flues of both boilers, square feet	628.32
Aggregate heating surface in the back smoke connections of both boilers, square feet	304.00
Aggregate heating surface in the upper group of flues of both boilers, square feet	1,262.92
Aggregate heating surface in the uptakes of both boilers, square feet	112.76

Total water heating surface in both boilers, square feet	2,648.00
Aggregate cross area for draft on the bridge walls in both boilers, square feet	18.16
Aggregate cross area for draft through the flues of the lower group of both boilers, square feet	15 97
Aggregate cross area for draft through the flues of the upper group of both boilers, square feet	14.14
Cross area of the chimney, square feet	14.75
Height of the chimney above the grate surface, feet	46.
Inner diameter of the chimney, feet and inches	4–4
Total steam superheating surface in both boilers, square feet	250.
Distance traversed by the products of combustion from the center of the furnaces to their delivery into the uptake, feet and inches	36–6
Ratio of the water heating to the grate surface	28.095
Ratio of the steam superheating to the grate surface	2.652
Ratio of the grate surface to the cross area above the bridge walls	5.190
Ratio of the grate surface to the cross area of the flues of the lower group,	5.9c2
Ratio of the grate surface to the cross area of the flues of the upper group,	6.665
Ratio of the grate surface to the cross area of the chimney	6.390

SCREWS.

There are two screws of cast iron, duplicates in all respects. Each screw is operated by its own engine, independently of the other. One screw is placed on each side of the vessel beneath the counter, and as far aft as possible without projecting beyond the sternpost; the after side of the screw being in the same cross plane of the vessel as the after side of the sternpost. Each screw has four blades equispaced around its axis. The forward outer corner of the blades is quadrantally rounded with a radius of 17 inches; the after outer corner of the blades is similarly rounded with a radius of 16 inches. With the exception of these curvatures, the forward and after edges of the blades, viewed in projection on a plane parallel with their axis, are nearly parallel, the mean length being 24½ inches, from which the extremes do not vary more than 1¼ inches. The thickness of the blades at the fillet of the hub is 4¼ inches. The pitch is uniform throughout.

The following are the dimensions and proportions of each screw:

Diameter of the screw, feet	9.
Diameter of the hub, feet	1½
Pitch (uniform) of the screw, feet	14⅝

Number of blades	4
Extreme length of the screw, feet	2.14
Fraction used of the pitch	0.54616
Helicoidal area of the screw, square feet	43.79
Projected area of the screw on a plane at right angles to its axis, square feet	33.78

WEIGHTS OF THE MACHINERY.

ENGINES.

	Pounds.
Cast iron	87,507
Cast bronze	6,985
Wrought iron	43,499
Wrought steel	1,877
Brass tubes in condenser	4,972
Reversing gear complete	2,276
Circulating pump complete	2,600
Two donkey steam pumps	2,000
Floor plates, flanges, cast-iron exhaust pipes, bolts, nuts, etc	48,017
Water in condenser and pumps, lagging, paint, putty, etc	5,267
Total weight of motive engines	205,000
Or 91.518 tons.	

BOILERS.

	Pounds.
Wrought iron and steel in the shells of the two boilers	62,971
Wrought iron flues in the two boilers	18,425
Wrought iron braces in the two boilers	10,420
Rivets, socket bolts, manhole and handhole plates, furnace doors, etc,	14,618
Chimney, cap, and casing	3,599
Cast-iron grate bars, bearers, etc	6,750
Safety valves, stop valves, feed and blow valves, ash pans, floor plates, etc	14,394
Copper steam pipes, feed pipes and blow pipes, etc	3,458
Water in the two boilers	69,197
Hair felt, paint, putty, etc	6,168
Total weight of boilers and appurtenances	210,000
Or 93.750 tons.	

SCREWS.

	Pounds.
Two cast-iron screws, keys, etc	8,100
Or 3.616 tons.	

MISCELLANEOUS.

Fire irons, ash chute, tools, stores, etc., pounds	2,500

Total weight in engineer department, exclusive of coal, tons	190
Coal, tons	138
Total weight in engineer department, inclusive of coal, tons	328

PERFORMANCE.

The following table contains the data and results of the mean performance of the *Albatross* at sea in light breezes and light swell, under the average conditions of practice, the hull being clean and the rate of combustion as great as can be permanently maintained with anthracite burned by natural draft.

The steam pressures in the cylinders are the means obtained from several hundred indicator diagrams carefully taken from both ends of each cylinder under the conditions of speed of vessel, revolutions of the screws and consumption of coal, given.

The piston of the steam cylinder of the independent pumping engine that supplied the surface condenser with injection water made 85 double strokes per minute during the performance in the following table, and developed 7.25 indicated horses-power; but this power, though produced by steam from the main boilers, is not included in the indicated horses-power developed by the motive engines. If it be included, it will lessen by about 2 per centum the cost of that power in fuel.

Under the conditions of the performance, the combustion of one pound of crude anthracite in the furnaces of the boilers will vaporize 8 pounds of water.

Table containing the data and results of the mean performance at sea in light breezes and light swell of the United States Fish Commission's steamer "Albatross," under the average conditions of practice, 1883 to 1886:

VESSEL.

Vessel's draught of water in feet and inches	forward	10–6
	mean	11–6
	aft	12–6
Vessel's greatest immersed transverse section, in square feet		263.787
Vessel's displacement, in tons of 2,240 pounds		953.824
Immersed external or wetted surface of hull, in square feet		7,175.000

ENGINE.

Steam pressure in boilers, in pounds per square inch above the atmosphere.	52.8
Steam pressure in receivers, in pounds per square inch above the atmosphere.	7.0
Vacuum in condenser, in inches of mercury	24.0
Proportion of throttle-valve open	0.68

Number of double strokes of the pistons made per minute	73.
Atmospheric pressure, in pounds per square inch above zero	14.68
Fraction completed of the stroke of the pistons of the small cylinders when the steam was cut off	0.68
Fraction completed of the stroke of the pistons of the small cylinders when the steam was released	0.95
Fraction completed of the stroke of the pistons of the small cylinders when the steam was cushioned	0.90
Fraction completed of the stroke of the pistons of the large cylinders when the steam was cut off	0 55
Fraction completed of the stroke of the pistons of the large cylinders when the steam was released	0.90
Fraction completed of the stroke of the pistons of the large cylinders when the steam was cushioned	0.80
Number of times the steam was expanded	5.079

TEMPERATURES.

Temperature, in degrees Fahrenheit, of the atmosphere	63.5
Temperature, in degrees Fahrenheit, of the engine room	102.5
Temperature, in degrees Fahrenheit, of the injection water	63.0
Temperature, in degrees Fahrenheit, of the discharge water	73 0
Temperature, in degrees Fahrenheit, of the feed water	98.0

STEAM PRESSURES IN SMALL CYLINDERS PER INDICATOR.

Pressure on pistons of small cylinders at commencement of their stroke, in pounds per square inch above zero	64.38
Pressure on pistons of small cylinders at point of cutting off the steam, in pounds per square inch above zero	56.27
Pressure on pistons of small cylinders at the end of their stroke, in pounds per square inch above zero	40.45
Mean back pressure against pistons of small cylinders during their stroke, in pounds per square inch above zero	22.62
Back pressure against pistons of small cylinders at the point where the cushioning began, in pounds per square inch above zero	20.32
Indicated pressure on the pistons of the small cylinders, in pounds per square inch	33.14
Net pressure on the pistons of the small cylinders, in pounds per square inch	29.64
Total pressure on the pistons of the small cylinders, exclusive of cushioning, the back pressure being reduced in the ratio of the length of the stroke of the pistons to the fraction completed of the stroke when the steam was cushioned, in pounds per square inch above zero	52.94
Mean back pressure against the pistons of the small cylinders, exclusive of cushioning, during the remainder of their strokes, in pounds per square inch above zero	22.00
Mean pressure of the expanding steam alone after the closing of the cut-off, in pounds per square inch above zero	47.32

STEAM PRESSURES IN LARGE CYLINDERS PER INDICATOR.

Pressure on pistons of large cylinders at commencement of their stroke, in pounds per square inch above zero	22.10
Pressure on pistons of large cylinders at point of cutting off the steam, in pounds per square inch above zero	15.50
Pressure on pistons of large cylinders at the end of their stroke, in pounds per square inch above zero	9.30
Mean back pressure against pistons of large cylinders during their stroke, in pounds per square inch above zero	6.00
Back pressure against pistons of large cylinders at the point where the cushioning began, in pounds per square inch above zero	5.24
Indicated pressure on the pistons of the large cylinders, in pounds per square inch	9.00
Net pressure on the pistons of the large cylinders, in pounds per square inch	8.00
Total pressure on the annular surfaces remaining of the pistons of the large cylinders after deducting from the areas of those pistons the areas of the pistons of the small cylinders; the back pressure being reduced in the ratio of the length of the stroke of the pistons to the fraction completed of the stroke when the cushioning of the steam began, in pounds per square inch above zero	13.32
Mean back pressure against the pistons of the large cylinders, exclusive of cushioning, during the remainder of their strokes, in pounds per square inch above zero	5.40
Mean pressure of the expanding steam on the annular surfaces remaining of the pistons of the large cylinders after deducting from the areas of those pistons the areas of the pistons of the small cylinders, and during the entire stroke of the pistons, in pounds per square inch above zero	15.00

HORSES-POWER.

Indicated horses-power developed in the small cylinders	183.959
Net horses power developed in the small cylinders	164.531
Total horses-power developed in the small cylinders	293.869
Total horses-power developed in the small cylinders by the expanding steam alone	84.055
Indicated horses power developed in the large cylinders	179.801
Net horses-power developed in the large cylinders	159.823
Total horses-power developed in the large cylinders, by the annular surfaces of their pistons remaining after deducting from the areas of these pistons the areas of the pistons of the small cylinders	192.166
Total horses-power developed in the large cylinders by the expanding steam acting upon the annular surfaces of their pistons remaining after deducting from the areas of those pistons the areas of the pistons of the small cylinders	216.403
Aggregate indicated horses-power developed by the engines	363.760
Aggregate net horses-power developed by the engines	324.354
Aggregate total horses-power developed by the engines	486.035

WEIGHT OF STEAM ACCOUNTED FOR BY THE INDICATOR.

Number of pounds of steam present per hour in the small cylinders at the point of cutting off the steam, calculated from the pressure there	7,230.085
Number of pounds of steam present per hour in the small cylinders at the end of the stroke of their pistons, calculated from the pressure there	7,557.435
Number of pounds of steam condensed per hour in the small cylinders to furnish the heat transmuted into the total horses-power developed by the expanding steam alone in those cylinders	525.413
Sum of the two immediately preceding quantities	8,082.848
Number of pounds of steam present per hour in the large cylinders at the end of the stroke of their pistons, calculated from the pressure there	6,151.365
Number of pounds of steam condensed per hour in the small and large cylinders and in the receiver to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder	1,088.019
Sum of the two immediately preceding quantities	7,239 383

EQUIVALENT PRESSURES ON THE PISTONS OF THE LARGE CYLINDERS.

Indicated pressure in pounds per square inch that would be on the pistons of the large cylinders were the indicated pressure on the pistons of the small cylinders divided by the ratio of the area of the small to that of the large cylinder, and the quotient added to the experimental indicated pressure on the pistons of the large cylinders	18.209
Net pressure in pounds per square inch that would be on the pistons of the large cylinders similarly calculated to the immediately above	16.209
Total pressure in pounds per square inch above zero that would be on the pistons of the large cylinders corresponding to the total horses-power developed by the engines	24.329
Back pressure in pounds per square inch above zero that would be against the pistons of the large cylinders corresponding to their experimental back pressure plus the difference in pounds per square inch between the back pressure, exclusive of the cushioning, against the pistons of the small cylinders and the mean pressure between the commencement of the stroke of the pistons of the large cylinders and the point of cutting off in the large cylinders	6.119

RATE OF COMBUSTION.

Number of pounds of anthracite consumed per hour	1,134.
Number of pounds of refuse from the anthracite in ash, clinker, dust, etc	189.
Number of pounds of combustible or gasifiable portion of the anthracite consumed per hour	945.
Number of pounds of anthracite consumed per hour per square foot of grate surface	12.032
Number of pounds of combustible consumed per hour per square foot of grate surface	10.026

Fraction of a pound of anthracite consumed per hour per square foot of heating surface	0.428
Fraction of a pound of combustible consumed per hour per square foot of heating surface	0.357
Per centum of the anthracite in refuse of ash, clinker, dust, etc	16⅔

ECONOMIC RESULTS.

Number of pounds of anthracite consumed per hour, per indicated horse-power	3.117
Number of pounds of anthracite consumed per hour, per net horse-power	3.496
Number of pounds of anthracite consumed per hour, per total horse-power	2.333
Number of pounds of combustible consumed per hour per indicated horse-power	2.598
Number of pounds of combustible consumed per hour per net horse-power	2.913
Number of pounds of combustible consumed per hour per total horse-power,	1.944

SPEED.

Speed of the vessel per hour in geographical miles of 6,086 feet	9.3750
Slip of the screw in per centum of its speed (pitch 14 feet 10 inches)	14.5207
Speed of the screw per hour in geographical miles of 6,086 feet	10.9678
Slip of the screw per hour in geographical miles of 6,086 feet	1.5928

DISTRIBUTION OF THE POWER.

In the following calculation of the distribution of the indicated horses-power developed by the engines, there is first deducted from that power the horses-power required to work the engines *per se* or unloaded. Then from the remainder, called the net horses-power applied to the crankpins, there are deducted the horses-power absorbed by the friction of the load, calculated at 7½ per centum of the net horses-power; and the horses-power required to overcome the surface resistance of the screw blades to the water, calculated for a resistance of 0.45 pound per square foot of helicoidal surface moving in the water with the velocity of 10 feet per second, increased in the ratio of the square of the experimental velocity to the square of 10; and the new remainder is divided between the power expended in the slip of the screw and in the propulsion of the vessel, in the ratio of the speed of the slip to the speed of the vessel.

	Horses-power.	Per centum of the net horses power.
Indicated horses-power developed by the engines	363 760	
Horses-power required to work the engines and shafting, *per se*	39.406	
Net horses-power applied to the crankpins	324.354	or 100 0000
Horses power absorbed by the friction of the load	24.327	7.5000
Horses-power expended in overcoming the surface resistance of the screw blades to the water	38.158	11.7643
Horses-power expended in the slip of the screw	38 025	11.7234
Horses power expended in the propulsion of the vessel	223.844	69 0123
Totals	324.354	or 100.0000

THRUST OF THE SCREWS.

The thrust of the two screws during the foregoing performance of the vessel, can be calculated from the speed of the vessel and the data in the above distribution of the power. The horses-power required to propel the vessel is therein found to be 223.844234, equal to (223.844234 × 33,000 =) 7,386,859.722 foot-pounds of work per minute. The speed of the vessel was 9.375 geographical miles of 6,086 feet per hour, or $\left(\frac{9.375 \times 6{,}086}{60} = \right)$ 950.9375 feet per minute; consequently, the resistance of the vessel at this speed, or its equivalent, the aggregate thrust of the screws, was $\left(\frac{7{,}386{,}859.722}{950.9375} = \right)$ 7,767.976 pounds.

The foregoing data and results have been compiled by the writer from the notes furnished to him by Chief Engineer Geo. W. Baird, U. S. Navy, under whose superintendence the machinery of the *Albatross* was constructed, and who was her chief engineer for several years afterwards. The performance given was made with the original flue boilers. After they were removed, horizontal fire-tube boilers with the tubes returned above the furnaces were designed by Chief Engineer Baird and substituted with the most satisfactory results, the speed of the vessel and the economy of the fuel being thereby considerably

increased, while the space and weight saved were of the greatest value for the purposes of the vessel.

Chief Engineer Baird's notes were copious and accurate; his observations of the vessel's speed, consumption of coal, power developed, etc., can be implicitly relied on. The indicator diagrams were carefully taken and very numerous, corresponding to the other data.

Among the many interesting experiments made by Chief Engineer Baird on board the *Albatross* was one determining the cost of the electric light of that vessel in foot-pounds of work done per unit of time.

The experiment was made by taking indicator diagrams from the cylinder of the dynamo engine when its shaft was making 300 revolutions per minute, computing from the mean indicator pressure, and subtracting from that quantity the pressure required to overcome the friction of the moving parts of the engine when the latter was unloaded, a friction due alone to the weight of these parts and to the quality of the lubrication, the remainder being the net pressure applied to the crank pin of the engine. From this latter pressure there is then subtracted $7\frac{1}{2}$ per centum of itself for the pressure required to overcome the friction of the load, the remainder being the pressure applied to the generation of electricity.

The friction of the load depends, of course, on the quality of the lubrication, like the friction of the moving parts, *per se*, of the engine. The latter friction can be measured directly by indicator diagram taken from the cylinder when the engine is unloaded, but the former friction can only be determined by independent experiment on the friction of well-lubricated metallic surfaces loaded and moving on each other.

The dynamo engine consisted of a single direct-action cylinder fitted with piston valves cutting off the steam by lap, and having the following dimensions:

Number of cylinders	1
Diameter of the cylinder, inches	$8\frac{1}{2}$
Diameter of the piston rod, inches	$1\frac{3}{8}$
Stroke of the piston, inches	10
Net area of the piston, exclusive of area of piston rod, square inches	56.0027
Space displacement of the piston per stroke, cubic foot	0.324

This engine was driven at the uniform speed of 300 double strokes of piston, or revolutions of the shaft, per minute. The reciprocating speed of the piston could not be allowed to fall lower, but the resistance overcome by it could be made variable so as to maintain that speed under all conditions, which was necessary to the maintenance of the light.

The electric light in use during the experiment consisted of 51 Edison house lights, each being of 16 standard candles power and equivalent to the light from 5 cubic feet of gas consumed per hour in an ordinary burner.

Thirty-one indicator double diagrams were taken from the steam cylinder, and the mean pressure from all was accepted as the indicator pressure exerted during the experiment, the corresponding reciprocating speed of piston being 300 double strokes per minute.

The results obtained by Chief Engineer Baird were as follows:

Number of double strokes of piston made per minute	300.
Mean indicated pressure in pounds per square inch on the piston	7.32
Pressure in fractions of a pound per square inch of piston required to overcome the friction of the working parts, *per se*, of the engine	0.82
Net pressure on the piston in pounds per square inch	6.50
Pressure in fractions of a pound per square inch of piston required to overcome the friction of the load ($6.5 \times 0.075 =$)	0.4875
Pressure in pounds per square inch of piston applied to the production of the electric light ($6.5 - 0.4875 =$)	6.0125
Number of foot-pounds of work applied per minute to the production of the electric light	168,358.1229
Number of foot-pounds of work applied per minute to the production of one electric Edison house light	3,301.1366
Number of foot-pounds of work applied per minute to the production of a quantity of electric light equal to that of one standard sperm candle	206.3212
Number of indicated horses-power developed by the engine	6 2112
Number of net horses-power developed by the engine	5.5154
Number of horses-power applied to the production of the electric light	5.1017
Number of horses-power consumed by the friction of the mechanism	1.1094
Per centum which the horses-power, applied to the production of the electric light, is of the indicated horses-power developed by the engine	82.138

Number of electric Edison house lamps maintained per indicated horse-power developed by the engine	8 211
Number of electric Edison house lamps maintained per net horse-power developed by the engine	9.247
Number of electric Edison house lamps maintained per hour by one horse-power applied to the generation of electricity for the same	9 997
Number of standard sperm candles required to give the quantity of light equivalent to the electric light produced by one horse-power applied to the generation of electricity for the same	159.9488
Number of cubic feet of illuminating gas required to give the quantity of light equivalent to the electric light produced by one horse power applied to the generation of electricity for the same	49 9827

Assuming one pound of coal to produce when retorted 5 cubic feet of illuminating gas, which, when consumed in an ordinary burner, would generate the light of 16 standard sperm candles; and assuming that the best triple-expansion steam engine worked at the highest practicable pressure and using its steam with the maximum economy obtainable from expansion, produces the horse-power applicable to the generation of electricity for $1\frac{1}{2}$ pounds of coal consumed per hour (equivalent to about $1\frac{1}{3}$ pounds of coal consumed per hour per indicated horse-power developed by the engine), then equal quantities of light would be obtained for 1.5 pounds of coal by the use of electricity, and for 1 pound of coal by the use of illuminating gas. But the commercial value of the bye products recovered in the gas manufacture, namely $\frac{5}{8}$ of a pound of coke for each pound of coal, besides the ammonia water and liquid hydro carbons per pound of coal, is so considerable as to reduce the coal cost of 5 cubic feet of illuminating gas to about one-third of a pound, making the coal cost of equal quantities of light generated by electricity and by gas in the ratio of $1\frac{1}{2}$ for the former and $\frac{1}{3}$ for the latter; that is to say, the same quantity of light generated by electricity costs in coal value $4\frac{1}{2}$ times as much as when generated by illuminating gas. The money first cost and after maintenance of equal light-producing electrical and gas plants, are very much greater for the former than for the latter.

The gas production of one pound of coal given in the above, is barely the practical average obtained, while the power produc-

tion of one pound of coal is taken at a rarely equaled maximum. As a practical round result the cost of producing a given quantity of light by electricity may be taken as five times as much as by ordinary illuminating gas, according to the experimental data.

The foregoing computations suppose the usual mixture of ordinary bituminous and cannel coal for the production of gas. If water gas be used, there are no bye products, but the lower cost of the pea sized anthracite screenings used in that case more than compensates their value, otherwise water gas would not be generated by preference.

The philosophical conclusion is that a very little electricity is the eqivalent of a great deal of heat, when the conversion is produced through the instrumentality of a steam engine.

AIR PUMPS.

By Passed Assistant Engineer F. H. Bailey, U. S. Navy.

In the maintenance of a good vacuum, the air pump is but one of three factors, the other two being air leakage and condensation. Of these three factors, air leakage is of quite as much importance as the size or style of the air pump; for, as will be seen later, it really determines the size of the pump. The utmost care should be taken to keep all stuffing boxes and valves about a condenser through which air may leak as tight as possible. These are properly details which belong to the management of the machinery. The designer can go no farther than to reduce the number of drains, auxiliary exhaust pipes, etc., connecting with the main condenser, to a minimum.

In designing the condenser the exhaust steam should enter at the top and as far away, measuring in the course which the steam has to travel from the air-pump nozzle, as possible. If necessary, baffle plates should be freely used to make the exhaust sweep over every part of the condensing surface. By following this plan all the cooling surface is rendered effective, and the air which enters with the steam accumulates to the greatest extent where it is being pumped out. An ideal condenser would be a pipe like a keel condenser, for here, as the steam and air enter at one end of the pipe and flow along, the steam is condensed, leaving an ever increasing percentage of air until the pump is reached, when we have the maximum percentage of air and the minimum of steam.

The pressure inside of a condenser is practically uniform, and is indicated by the vacuum gauge. It is the sum of the pressure of the air and the pressure of the steam. The pressure of the steam is always that due to its temperature. We can, therefore, analyze the vapor in the condenser at any point by means of the

vacuum gauge, the thermometer and our steam tables. Thus, if the temperature of the vapor in the channel way of a condenser was 110 degrees and the absolute pressure 2.58 inches of mercury, we would know that no air was present; but if the pressure was 3.58 inches, we would know that there were 2.58 parts by volume of steam to one of air.

The principal office of the air pump is to prevent the water and air from accumulating in the condenser. The volume of water is so small in comparison with the volume of air to be removed from the condenser, that it is not taken into consideration in determining the required displacement of the air-pump piston per minute. This is shown by the following example: Taking the capacity of our air pumps at .2 of a cubic foot per I.H.P. per minute, and assuming that our engines use 15.5 pounds of water, or $\frac{1}{4}$ of a cubic foot per hour, we have $\frac{1}{4} \times \frac{1}{60} = \frac{1}{240}$ of a cubic foot per minute, only enough to fill $\frac{1}{48}$ of our cylinder each stroke.

We gain nothing by pumping the steam out of our condenser. It would be better for it to remain and be condensed, but it is mixed with the air, and incidentally we pump some of it out along with the air. The amount of air depends so largely on leakage, which in turn depends on the care with which the machinery is looked after, that the actual capacity of a pump necessary to produce a given vacuum would vary from day to day just as the leakages varied. Now, here is where the independent air pump shows one of its several advantages. We can vary its speed as we wish to suit the necessities of the case. When the pump is attached to the engine it should be made of sufficient size to give a fair vacuum at the ordinary speed of the engines. It will then be found to be larger than necessary for high speeds, and too small at low speeds. This is owing to the fact that many of the air leaks increase as the steam pressure falls. Some air is, of course, pumped in with the feed water, but the great mass of air, where surface condensers and float valves on the feed suctions are used, comes through leaks.

A good average value for independent air pumps for men-of-war is .2 of a cubic foot displacement of air-pump piston per minute for each I.H.P. When the pump is attached to the engine

this capacity should be increased about 50 per cent. for full power. In actual practice some pumps are made with double this capacity, while others have only about one-half as much. As we shall presently see, the size may be varied considerably without making a very marked change in the vacuum produced. With careful attention to air leaks much smaller pumps may be used and good results obtained, while with leaky piston rod stuffing boxes much larger pumps will not give a good vacuum.

The foot valves are raised by the pressure of the air, water and steam in the condenser. Let us see how much pressure we have to do this. If there were no clearance we would have the whole pressure in the condenser to raise these valves; but no air pump is made without clearance, and many of them have large clearances. In well designed pumps this clearance is filled with water, and we will assume that it is in this case. When the piston begins to move, a partial vacuum is formed behind it, the amount of which for one thing depends on the temperature of the water, since the water will boil and fill this space with steam due to its own temperature. To illustrate this I have worked out an example. Assume the clearance at 10 per cent. and the displacement of the piston at 1 cubic foot, and the temperature of the water filling the clearance spaces at 120°. The water will weigh 6.25 pounds. When the piston moves to the other end of the stroke (supposing the foot valves to remain closed) some of the water will evaporate, filling the space behind the piston with steam. We will have 1 cubic foot of steam, and a certain amount of water of a somewhat lower temperature than before, but the water and steam together will have the same total weight and heat as before. A cubic foot of steam at this temperature contains about 5 thermal units of latent heat. We have therefore reduced the temperature of the water by about $5 \div 6.25 = .8$ of a degree, which corresponds to a reduction of pressure of less than .04 of a pound, or .08 of an inch of mercury per square inch. In other words, if no air were present there would be less than .04 of a pound pressure per square inch to lift the valves when the vacuum gauge showed an absolute pressure of 1.685 pounds per square inch, or a vacuum of

26.6. The water would not help to raise the valves until enough had accumulated behind them to exert a sensible pressure. This could only occur when, for some reason, the valves had failed to lift for a number of strokes. If air sufficient to produce a pressure of one inch of mercury had been present in the condenser, and the temperature had remained the same, we would have had just so much more pressure operating to open the foot valves. The vacuum would have shown less than 26.6 by just one inch, or it would have read 25.6. A 26 inch vacuum is not unusual, and this shows how little force we have to lift our foot valves when we have a good vacuum—only about .7 of an inch of mercury. With the same temperature and a 24-inch vacuum, we would have about 2.7 inches of pressure, or about four times as much as before.

Our foot valves having opened, the cylinder will fill with water, air and steam, the total pressure being less than that in the condenser by the loss due to friction, and the pressure required to lift the foot valves. As the piston begins to return and compress the contents of the cylinder, the water, being incompressible, retains the same volume until the delivery valves raise. The steam remains at the pressure due to its temperature, and the volume of water in the cylinder is so great that the temperature can not rise much, so that the steam is almost entirely condensed without increasing its pressure materially. The air will be steadily compressed as the piston returns, the pressure rising according to Mariotte's law. If we had at the begininng of the return stroke sufficient air present to produce a pressure of 1 inch of mercury and a water temperature to produce a pressure of 3 inches, and the discharge valves lifted at 30 inches, the delivery valves would raise when the space not occupied by water had been reduced to $\frac{1}{27}$ of its original volume, and there would be discharged, assuming all the clearance space filled with water, $\frac{1}{27}$ of the original volume of air at a pressure of 27 inches and $\frac{1}{27}$ of the original volume of steam at a pressure of 3 inches of mercury. This is based on the assumption that the temperature remains constant, and this is true in the great majority of cases, although there may be some cases where the water has failed to

keep the temperature, and consequently the pressure, from rising faster than this assumption would indicate. Our pressure would vary from 4 inches at the beginning of the stroke to 30 at the end, an increase of 750 per cent. This is shown on the appended card (A), where the two broken lines represent the water and steam respectively, the dotted line the air, and the full line the sum of the air, water and steam.

In practice, our pumps do have more or less clearance not filled with water. Its effect is to reduce the efficiency of the pump, or the amount of vacuum which the pump could produce:—thus with no clearance, the limit would be a perfect vacuum; with the clearance filled with water it would be that due to the temperature of the water. With a clearance equal to C, a volume at the beginning of the stroke equal to V, the pressure of the steam due to its temperature p, and the total pressure at discharge P, the limit of the vacuum produced will be $\frac{(P-p)\,C}{V} + p$. For example, let the clearance $C = \frac{1}{10}\,V$, and the pressure of the steam be 3 inches of mercury, and the pressure of the discharge 20 pounds absolute, or 40 inches of mercury, the limit of the vacuum which we could produce would be $\frac{(40-3)\,V}{10\,V} + 3 = 6.7$, or 23.3 per gauge. A single-acting air pump with foot, bucket and delivery valves as commonly used would do very much better, as the above would represent only that part of the pump between the bucket and delivery valves. The action of the part between the foot and bucket valves would be the same, only the pressure of discharge would be very much lower, rendering the effect of clearance very much less. Allowing the same ratio of clearance as above, we would have for that part of the pump between the the foot and bucket valves $\frac{(6.7-3)}{10\,V} + 3 = 3.37$ absolute or 26.63 by gauge. It will be noticed that in this case the limit is only .37 of an inch of what could be obtained without any clearance. It also shows why so much better results are often obtained when double-acting pumps are converted into single-acting ones, notwithstanding the fact that their capacity is reduced one-half.

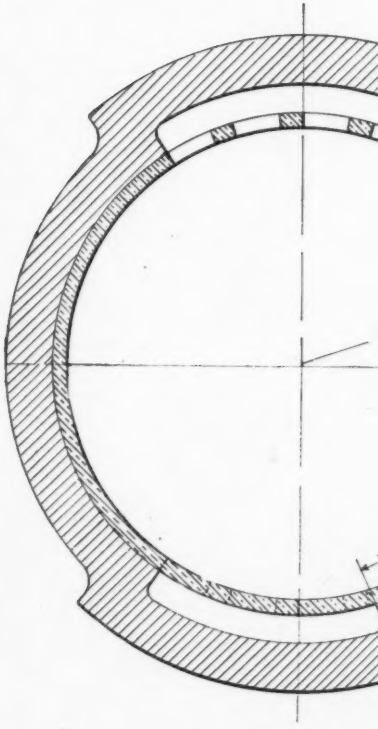
As air and water are all we care to pump out—in fact, the most of the steam becomes water before it is pumped out—water makes a very good filling for the clearance spaces, but the above shows how necessary it is to avoid all pockets, however small, where air can collect without being expelled at each stroke.

We cannot build our pumps without clearance, and in trying to fill these spaces with water the speed becomes important; for when a certain speed is reached the water begins to get churned up or mixed with air bubbles, and of course no longer makes a solid filling for the clearance. In high-speed pumps the water is undoubtedly thrown from the piston of the pump as the piston is retarded by the action of the crank near the end of the stroke. This causes more or less of the water to be thrown out through the delivery valves, instead of remaining to fill up the clearance of the pumps. The clearance spaces are more easily kept filled with water in vertical pumps than in horizontal ones, and since clearance space is less harmful in single-acting pumps than in double-acting ones, high-speed pumps should be made single acting and vertical where possible. The speed at which a pump can be run and give good results depends somewhat on the design, but the efficiency of a double-acting pump may be expected to begin falling off when a piston speed of from 100 to 200 feet, depending on the length of the stroke, is exceeded. Such pumps are often attached directly to the engine, when, of course, they have very much higher piston speeds, generally the same as the steam piston, but in such cases they do not usually give very good vacuum. Vertical single-acting pumps can be run considerably faster. A piston speed of from 400 to 500 feet should give fair results.

It is impossible to lay down any hard and fast rule fixing the size of an air pump. It depends on air leakage, which is an uncertain factor. Fortunately for us, however, the size can be varied greatly without making a very great change in the vacuum. A good air pump should maintain a vacuum within one inch of that due to the temperature of the condenser; or with the temperature of the air-pump discharge, which is generally taken instead of the condenser, of 120°, 25.6 would be a good fair

vacuum. Probably the efficiency of a pump working with as good a vacuum as this would not be more than 50 per cent., as it would take about a half inch of air to raise the valves and to overcome friction. Doubling the size of the pump would diminish the pressure of air in the pump, or in the condenser, above that required to lift the valves just one-half. Since we require a half inch to raise the valves, the vacuum would be increased but ¼ inch. In a similar manner it can be shown that with a pump of only half the capacity our vacuum would fall off but half an inch. The annexed table of air data shows how little the mere size of the pump has to do with the vacuum, provided the pump is not very much too small. Too large a pump is more objectionable than too small a one, for not only are they heavy, occupy more space and take more steam, but they are more irregular in their action. We have already seen how little force we have to lift our foot valves, and the better the vacuum the less force we have. If our pump is too large there may be times when these valves will not lift for several strokes, allowing the channel ways to fill up with water, and then when the pump does act it is flooded. When doing no work it speeds up, and when it begins to work again it is flooded with water, slows down and may even stop. A capacity of .2 of a cubic foot per minute for each indicated horse power on the full-power trial is ample for vertical single-acting pumps. A capacity of one-third this has given good results on one of our largest ships.

The area through the bucket valves should be made as large as possible, as these pumps may have to pass almost solid water. The velocity of flow ought not to exceed 1,000 feet per minute. If the bucket is not large enough to admit of so low a velocity, then the stroke should be shortened and the diameter increased. The foot valves should have the same area as the bucket valves, but the delivery valves may have a larger area with advantage. Pumps are almost always broken, when they do break, from too much water, so that as free an exit as possible should be provided for the water. Circular valves are the best. They should be from 4 to 7 inches in diameter. They should be made as light as possible and held down to their seats with a conical spring of



Section on

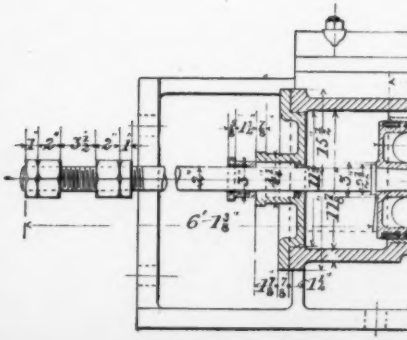
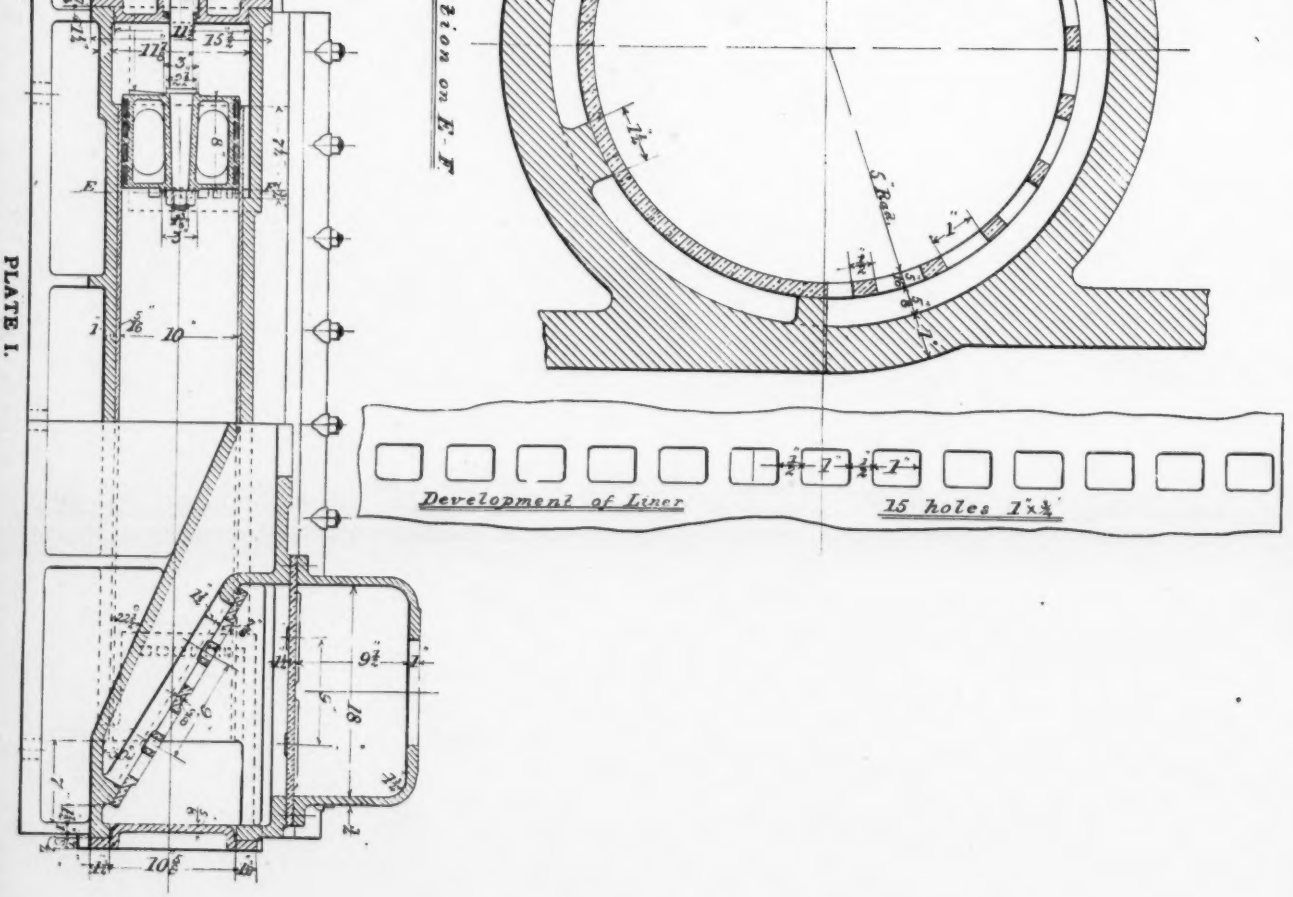


PLATE I.



AIR PUMP
for

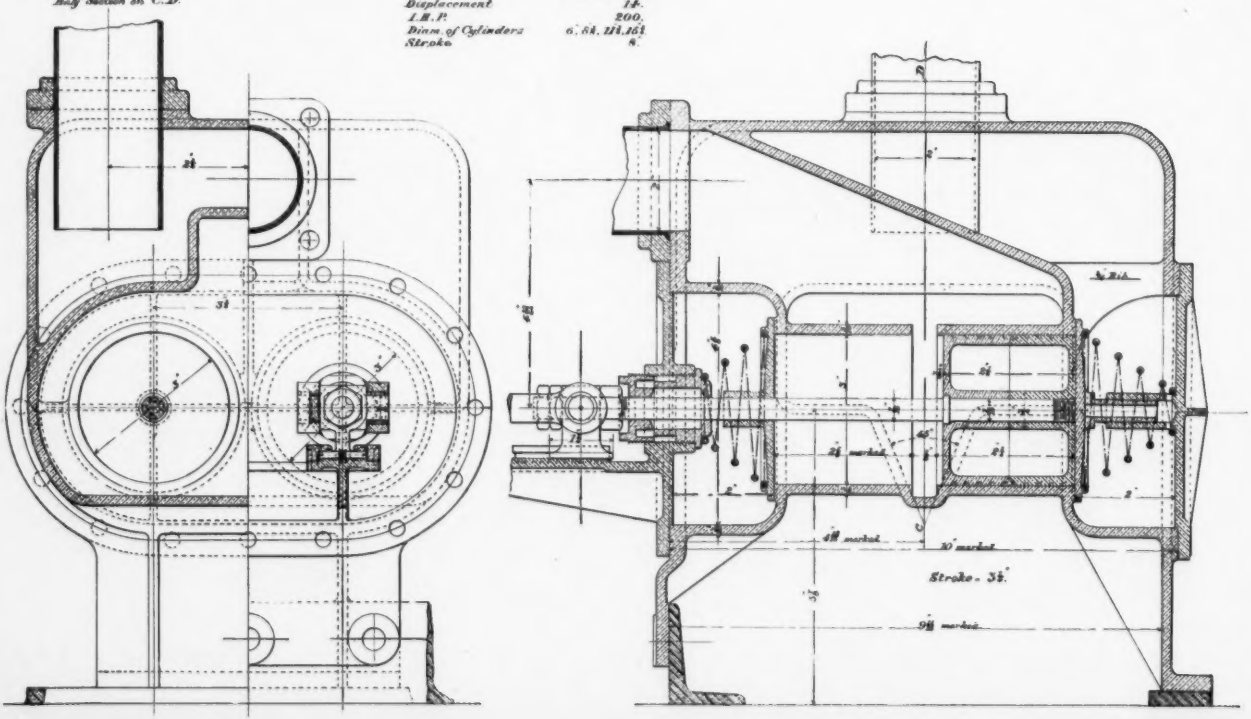
PLATE II.

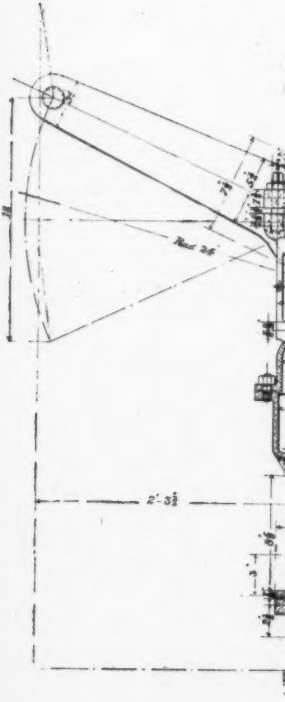
AIR PUMP
for
3½ Class Torpedo Boat for U.S.S. Maine.

PLATE II.

Single Screw	One Engine.
Displacement	14.
I.H.P.	200.
Diam. of Cylinders	6. 8¼. 13¼. 16½.
Stroke	8.

Half Section on C.D.





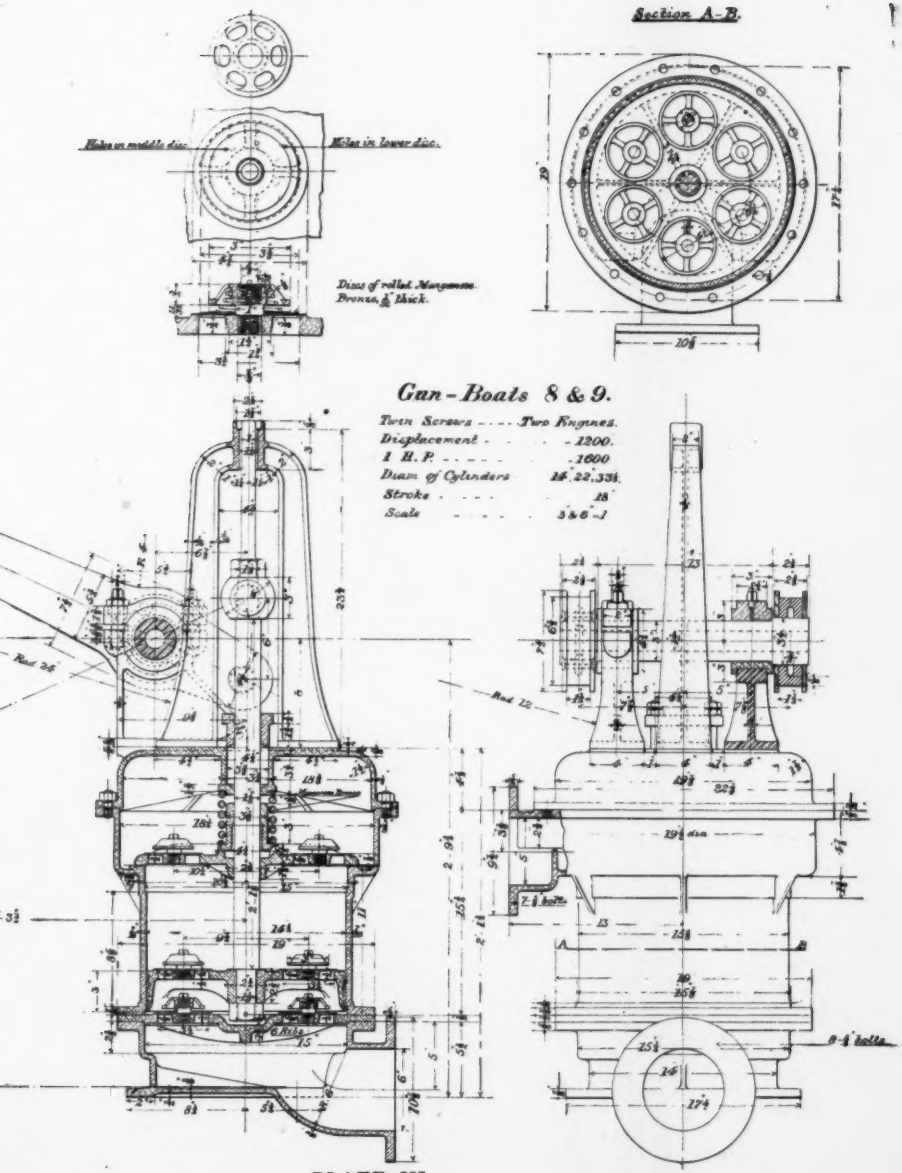

Section A-B.
Holes in middle disc.
Holes in lower disc.
Discs of rolled Manganese Bronze, ⅝ thick.
Gun-Boats 8 & 9.
Twin Screws - - - - Two Engines.
Displacement - - - 1200.
I. H.P. - - - - - 1600
Diam of Cylinders 18. 22. 33½.
Stroke - - - - 18
Scale - - - - 3 & 6 - 1
Rod 26
Rod 12
19½ dia
8-1¼ bolts
6 Bolts

PLATE III.

brass wire. Hard rubber is a very satisfactory material for valves, and is largely used for that purpose. Sheet manganese or phosphorus bronze also makes very good valves. When made of this material each valve is made of three thicknesses of different diameters, the smaller being put on top. The middle and upper sheets are perforated to allow for water lubrication. Heavier cast valves have been tried, but the result was far from satisfactory. The lift of the valve is important at high speed, and should vary from $\frac{1}{4}$ to $\frac{5}{8}$ of an inch. Too large a lift prevents the valves from closing quickly and is likely to make them pound at high speeds.

Professor I. N. Hollis, of Harvard University, who was formerly an assistant at the Bureau under Engineer-in-Chief Melville, has devised an improvement for double-acting air pumps which will undoubtedly double their efficiency at high speeds. It will be tried on the U. S. S. *Hartford* when she is re-engined. It consists in connecting the clearance spaces at each end of the pump with the barrel of the pump by means of ports cut through the liner. These ports are cut so that the piston overruns them at each end of its stroke by about half an inch. The effect is to put the two ends of the pump in communication and allow the pressures in the two ends to equalize before the piston returns. The pressure behind the piston is reduced to almost that of the condenser, so the foot valves begin to open almost as soon as the piston begins to return. It has the same effect as reducing the clearance. The *Hartford's* pump is shown on plate I.

Another scheme is to cut a port in the center of the barrel, which is always in communication with the condenser. The piston is made equal in length to the stroke less the width of the port, and overruns the port at each end of the stroke. This allows free communication at each end between the condenser and the pump so that the pump barrel is sure to be filled each stroke, and it does away with foot valves. As the piston returns it covers the port and imprisons all the contents of the cylinder, which are discharged at the end of the stroke. Clearance is reduced to a minimum by making the ends of the piston flat and the barrel of such a length that the piston just comes flush with

it. One large delivery valve covers each end of the cylinder. This pump is especially suitable for high speed and small sizes. The cylinder must fill each time, and if it should fill with water it can not break down, since the whole end of the pump comes off. In large sizes small valves of the usual pattern should be put in the ends. These valves would open under ordinary circumstances (having lighter springs) while the large valve would open in case there happened to be a larger quantity of water than usual present. This style of pump is now being used on our steam cutters, and will also be used on the torpedo boats for the *Maine* and the *Texas*. (Plate II.)

In a comparison of independent air pumps three things should be considered:

1st. The weight and space which the pump occupies.

2d. The economy of the pump so far as its steam consumption is concerned.

3d. The reliability of the pump as a whole.

Weight and Space.—So far as the water end of the pump is concerned it matters little how the pump is driven, provided the speed is suitable; and for the purposes of comparison we may assume that the water end is exactly alike in all the pumps. This simplifies the subject and leaves us free to consider only the steam end of the pump.

There are two general methods in use for driving pumps, one is known as the direct-acting system, in which the valve gear is worked by tappets or some equivalent device, of which the Blake, Davidson and Dow are good examples. The other is known as the fly wheel system, in which the valves are driven by an eccentric in the ordinary fashion. Now we should make our pump so it can start in any position, and to do so we must use steam cylinders large enough for the steam pressure to overcome the maximum load on the air pump piston in all positions. To do this we must use cylinders of practically the same capacity in either case, whether our pump is driven direct by means of a beam, or by means of a crank and fly wheel with or without gearing. Now the fly-wheel pumps must have, in addition to their cylinders, a shaft and heavy fly wheel, and will also require

heavier framing. Gearing the pump will make the fly wheel smaller, but the extra weight of gearing will fully offset the saving of weight on the cylinders and fly wheel. The space required for a fly-wheel pump will also be greater than that of a direct-acting pump by at least the space occupied by the fly wheel. Weight and space are very valuable on ship board, but not so much so as economy of steam and reliability. As regards weight and space, the lightest pump will be the direct-acting pump, and it will probably represent an average saving of 30 per cent. in space, and from 30 to 40 per cent. in weight.

Economy.—Looking at one of the annexed air-pump cards, we will notice that nearly all the work is done near the end of the stroke, although if the pump happens to be flooded the card may be nearly a rectangle. A steam card is the very reverse of a pump card. In it the work is done at the beginning of the stroke, and near the end where the pump needs the most work there is the least; in fact the curve of unbalanced pressure would show a negative pressure. Now, we know of no system of cranks or levers whereby we can reverse either of these cards. If we did, the problem would be much simplified; but as it is we must resort to other methods. We usually adopt one of two, or a combination of two, principal ways. One method is to use a heavy fly wheel with one steam cylinder, or a lighter one in connection with several cylinders, each connected to a common shaft, the cranks of which are so arranged that one or more of the pistons is always acting on a crank favorably placed for producing a large turning moment. The pumps may be driven direct or by means of gearing. In the Navy we generally use two steam and two water cylinders, with fly wheels and without gearing. The other method is that used in direct-acting pumps, where, by wire-drawing and cushioning, or similar means, the pressure on the steam piston is modified to suit the pressure on the pump piston. The first double-cylinder fly-wheel pumps were not satisfactory, as it was impossible to run them slow without danger of their stopping. Hoping to get better results, geared pumps were designed, and two for the *Oregon* have been built. Gearing is right in theory, as we get high steam-piston speed with the proper speed of air-pump piston. In practice

they are noisy, cumbersome and heavy, and I think they will never come into general use. Increasing the number of cylinders adds to the smooth working of the pump, but it also adds to the complexity of the machine as a whole. A three-cylinder air pump is now in use on the *Olympia*, and it is said to work beautifully. I am inclined to believe, however, that two steam cylinders are enough for good, practical results. One cylinder can be made to do about as well, but it would require an enormously heavy fly wheel, and would have the further practical disadvantage of being hard to start if for any cause it were stopped while there was a vacuum in the condenser.

We will not endeavor to give a formula for the size of the fly wheel; it depends on too many things which the designer may change at will. The fly wheel should be heavy enough not only to counteract the irregularities of the work during a revolution, but to keep the pump from stopping when it becomes flooded with water. To produce as steady a motion as our best engines, would require fly wheels heavier than all the rest of the pump combined.

Let W = weight of fly wheel.
" R = radius of gyration.
" S = stroke of pump.
" A = area of pump piston.

Then $\frac{WR^2}{AS}$ is a measure of its dynamic force, or power to prevent irregular motion. In the case of the *Charleston*, we have:—

$W = 1400$; $R = 24$; $S = 10$, and $A = 452$; or the dynamic value is $\frac{1400 \times (576 = 24^2)}{452 \times 10} = 180$. If this value for the *Charleston* be represented by 100, then we would have the following values for some of the other ships:

Charleston, ...	100
Cincinnati and *Raleigh*, ...	46
Philadelphia, ...	24
Machias and *Castine*, ...	23
Detroit and *Montgomery*, ...	18
San Francisco, ...	11

The *Charleston's* air pump has one of the heaviest fly wheels of any of our pumps in proportion to its size. Owing to Mr. Hollis' careful observations we have more reliable data regarding that pump than any other. For this reason we have selected it for the purpose of determining how much variation of speed there would be when working under normal conditions, and giving a card similar to the one marked G.

The pump has two steam cylinders, each having a diameter of 10 inches, and two water cylinders of 24 inches diameter, with a common stroke of 16 inches. The fly wheel weighs about 1,900 pounds, of which 1,400 pounds is in the rim. The radius of gyration is about 24 inches. The cranks are placed nearly opposite. The cards marked K are from the water end when the pump was making about 70 revolutions per minute. The variation in speed, as calculated from these cards and the steam card already referred to, would be for a single revolution, from a speed of about 73 revolutions to 66¾ revolutions. It would not pass the centers at 29 revolutions. At 100 revolutions the variation would be less than 1 revolution from the average. We would expect a pump of this kind to stop occasionaly, when it happened to get an extra heavy load of water; and we believe it did until Mr. Hollis took it in hand and reset the valves, producing the cards marked I.

Now we want our air pumps to run with reasonable steadiness and at the same time we don't want to make a pump all fly wheel. The *Charleston* has one which is plenty heavy enough for ordinary running, and by sacrificing a little to economy, we can make a pump which will run at any speed without danger of stopping. On the *Charleston*, Mr. Hollis solved this by pushing the eccentric back so that steam would follow about full stroke. Now by using two cylinders and placing the cranks at any position not opposite or together, and making them large enough so that the total pressure on the steam piston will be greater than that on the pump piston, and also setting the valves to follow full stroke, the pump will run, and it will start from any position when the steam is turned on.

Now let us see what effect the inertia of the fly wheel and other

moving parts will have on the steam card and the economy of steam. We will assume that the speed is governed by a throttle. The space between the main valve and the throttle valve has a marked influence on the card and on the running of the pump. The same is also true of port area. Now it is plain that if the volume of steam between the throttle valve and piston is very large, and if there is no wire drawing, we will get the card marked B, which will be a plain rectangle whatever be the size of the fly wheel. With a heavy fly wheel the pump will run smoothly. With a light fly wheel the speed of the pump will be increasing until we reach the point *a* on the air pump card, when there will be more or less of a blow struck. With light moving parts, as in direct-acting pumps, or with those which have too small a fly wheel, this blow would be serious and might wreck the pump. This has occurred in one or more instances. Reducing the space between the throttle valve and the main valve to small dimensions, or, better yet, by throttling with the valve itself or through the steam port, and using a heavy fly wheel, we will get a card marked C. If, however, the throttling is done with the valve so that the opening varies with the speed of the piston, we would get an approximate rectangle as before. It will be noticed that in this card the steam pressures have varied with the speed of the piston, and are highest at the beginning and end of the stroke. With a light fly wheel we get the card marked D. Here the pressure varied with the motion of the piston, and the motion of the steam piston varied as the resistance to be overcome, or as the pressure on the air-pump piston. We have a high velocity of piston until the air pump begins to do its work, when the piston slows down and the steam pressure rises until at the end of the stroke it is a maximum. This card is much like the air-pump card, since it has the highest steam pressure at or near the end of the stroke. Throttling the exhaust, either in the port or at the valve, will give a somewhat different card, but the effect will be the same. We get a card like the one marked E. Now in these cards lies the whole secret of making a pump without a fly wheel, or with too light a one, run smoothly. The action is practically the same, except that in one case the valve

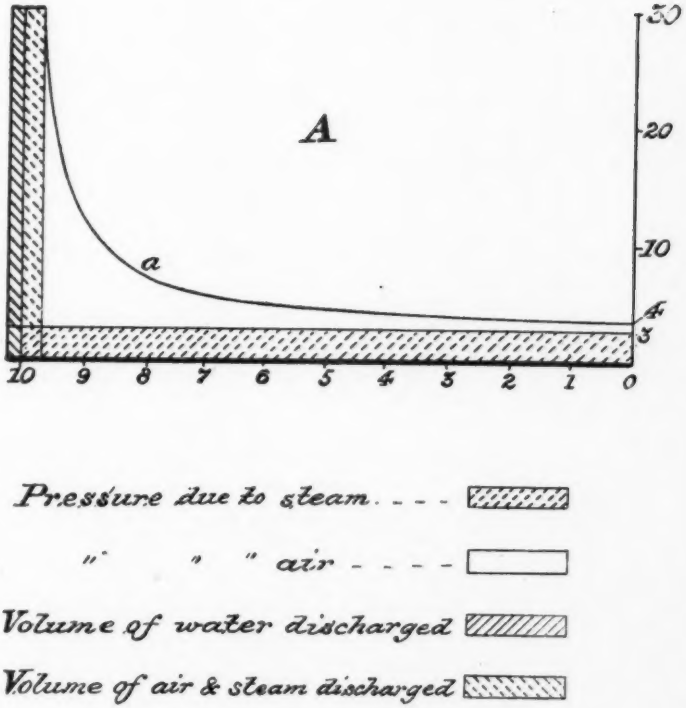
is moved with an eccentric, and in the other it is moved by an auxiliary steam piston.

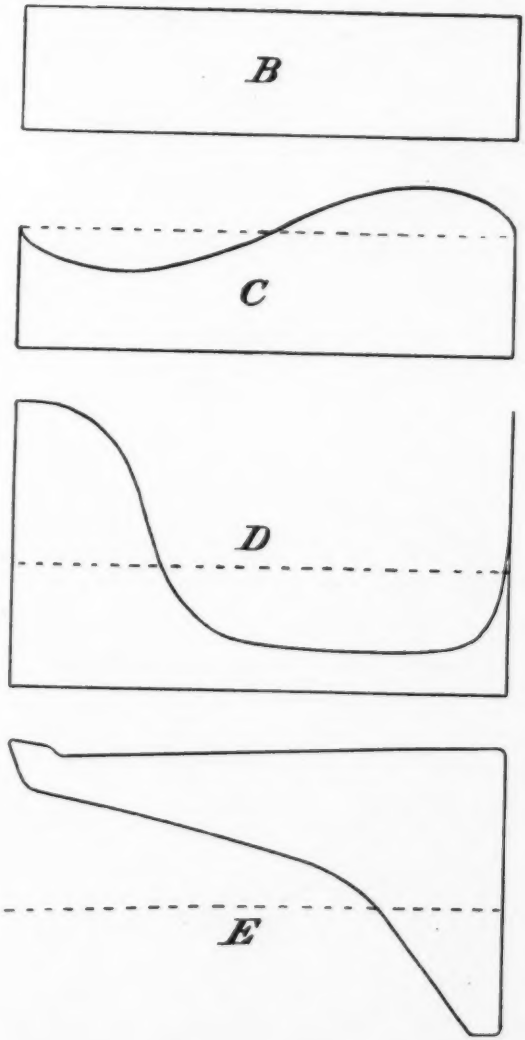
The mean steam pressure of all these cards should be the same, but the amount of steam used will be proportional to their terminal pressures. It will be seen at once that B and C used less steam than D and E. Since with heavy moving parts, or with a heavy fly wheel, we get the lowest terminal pressures, and as we lighten these parts the terminal pressure increases, it is clear that a fly-wheel pump will be more economical than a direct-acting one. The energy which in a direct-acting pump is wasted in wire-drawing the steam during the early part of the stroke, is here used to store up energy in the fly wheel to be used in the latter part of the stroke, to assist the lower steam pressure in completing the revolution. Adding on other pumps, as circulating pumps for instance, has much the same effect, and adds very greatly to the economy and smooth working of the pump when the moving parts are light. When the main engines are to run at some nearly constant speed, then the best thing to do, is to attach it to the main engines themselves; for here we have the most economical steam end, and plenty of mass to make it run steadily. Weights added to the ends of the beams in the *New York's* pump would increase the economy, but of course it would limit their range of speed. With a fly wheel this is different, as the range of speed is increased with the size of the wheel—or perhaps we should say the amount of stored energy which it contains.

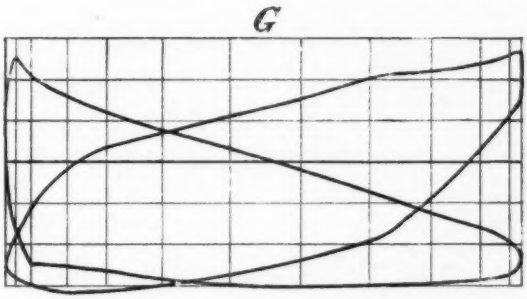
Regarding the size of ports. The air pumps for the *Machias* and *Castine* and for the *Detroit* and *Montgomery* gave considerable trouble when first tried. They would not run, except at high speed, without stopping. Their motion was very irregular, so much so that it was almost impossible to run them more than a few hours at a time. The attention of the Bureau was called to their behavior and suggestions for their improvements asked for. The pump was examined, and although of unusual design, there was no cause found to account for the irregular working of the pump. The fly wheels were found to be very light, but the space did not permit of much larger ones being used, so some

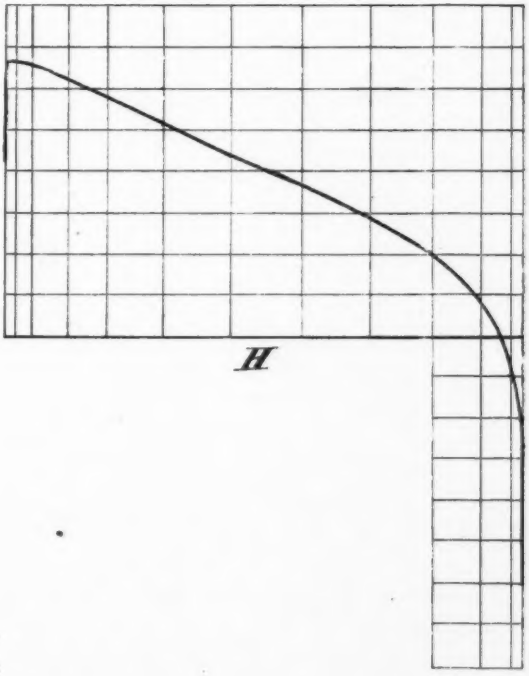
other means had to be tried to get a steadier motion. A study of the steam end showed that there were large steam chests and considerable length of pipe between the throttle valve and the steam chest; also that the steam ports had an area of about $\frac{1}{10}$ of the cylinder, and that the valves were designed to cut off at about $\frac{3}{4}$ stroke, and to have the usual amount of compression. There is no need of compression on the steam piston of an air pump, for the compression in the air pump itself is altogether too much for the purpose of smooth running; neither do we want a cut off with a light fly wheel, for if for any reason the pump slows down it may not be able to complete the stroke, through lack of pressure, and will stop. This was seen at once and it was decided to set the eccentrics with no angular advance. This was carried out as well as it could be on the *Montgomery* with the old eccentrics, and the pump was tried with the cranks at right angles, and also nearly opposite. The pump ran steadiest with the cranks nearly opposite, but would stop sometimes. With the cranks at right angles the motion was more jerky, but the pump did not stop.

We all know that a fly wheel tends to make a machine run steadier, and we do know that direct-acting pumps are made to run with reasonable steadiness without any fly wheel. Now, why couldn't we make this work the same way? Upon a comparison with direct-acting pumps, we found that our ports were about 4 or 5 times as large as was the usual practice in direct-acting pumps, so that there was no wire-drawing of steam or exhaust in the ports. The large space between the throttle and steam valve prevented the pressure from fluctuating rapidly, so that there was a fairly uniform pressure upon the steam piston. The next step was to devise a valve which should wire-draw the steam. This was accomplished by giving the valve so much lap that the steam port would be entirely covered all the time, and then cutting a square notch through this lap, so that the virtual opening of the valve was reduced as much as desired. As there is but little work done when the piston is descending the upper notch was made smaller than the lower one, so as to give a reasonably regular motion to the pump when the cranks


A
a
30
20
10
4
3
10
9
8
7
6
5
4
3
2
1
0
Pressure due to steam. - - -
" " " air - - - -
Volume of water discharged
Volume of air & steam discharged

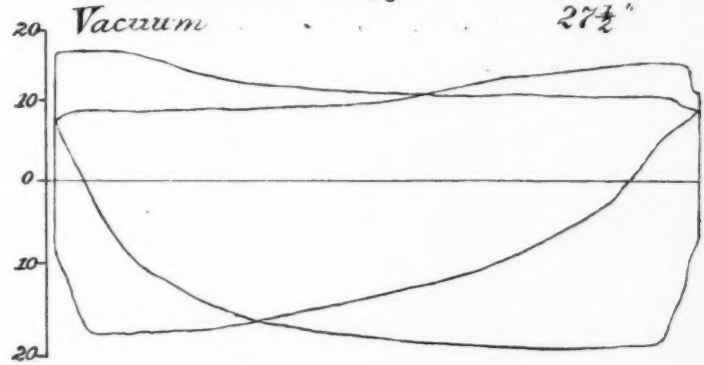

B
C
D
E


G

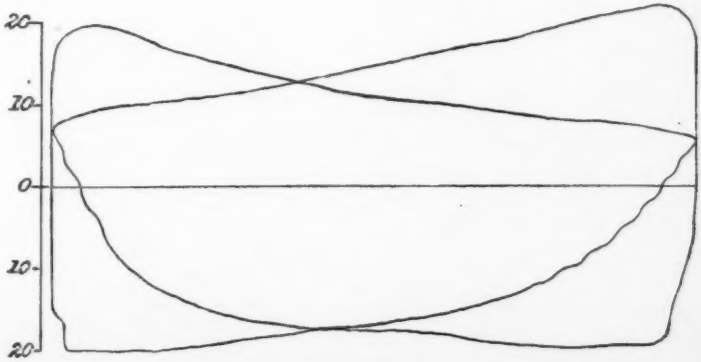

H

U. S. S. Charleston.

Date & Hour	Aug. 22d 6 A.M.
Revolutions per min.	71
Scale	20
Mean Pressure eff.	16.74
Vacuum	27½"

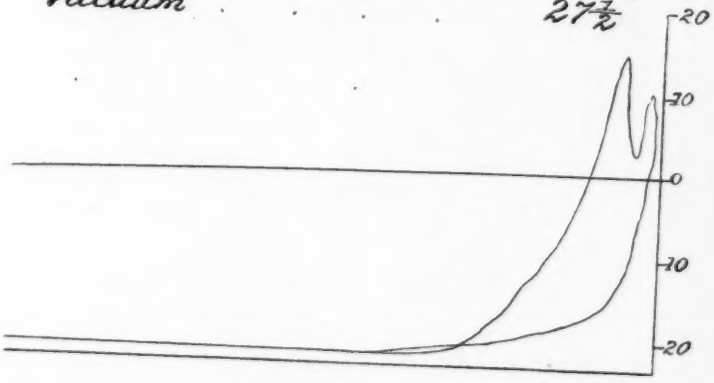


I

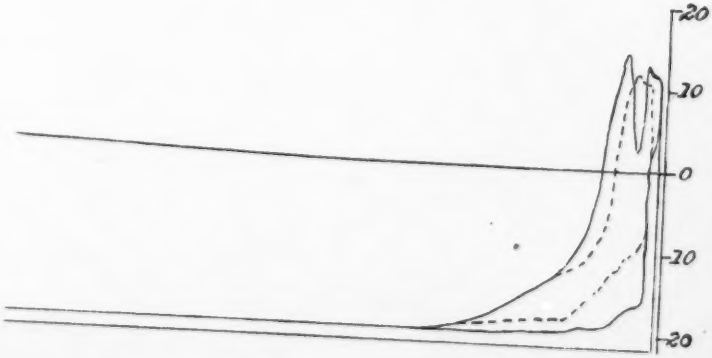


U. S. S. Charleston.

Date & hour . . . Aug. 22d 6 A.M.
Revolutions per min. . . 71
Scale 20
Mean Pressure, eff.
Vacuum . . . . . 27 1/2"



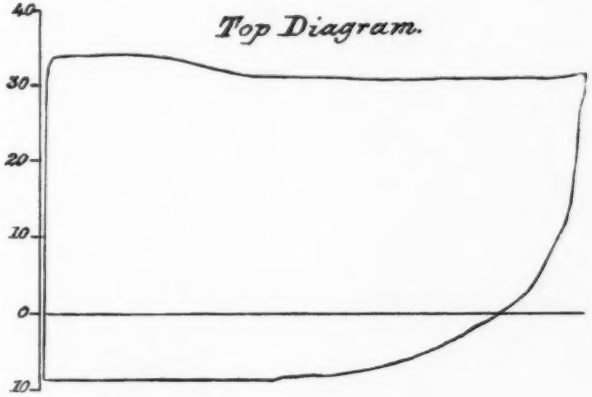
K



U. S. S. New York.

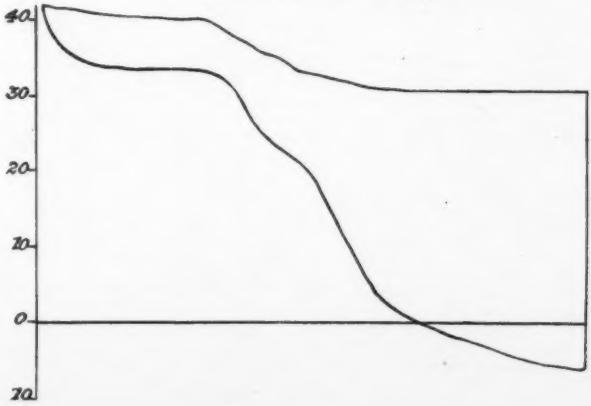
Date & hour	*Dec. 22/93 - 2 P.M.*
Revolutions per min.	*12.7*
Scale	*20*
Mean Pressure, eff.	*36.2*
Vacuum	*23*

Top Diagram.



Vacuum	*23*
Mean Pressure, eff.	*18.9*

Bottom Diagram.



U. S. S. New York.

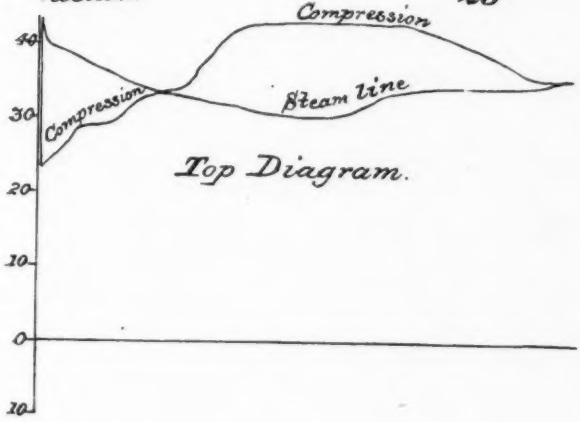
Date & hour

Revolutions per min. 12.27

Scale 20

Mean Pressure, eff. 42

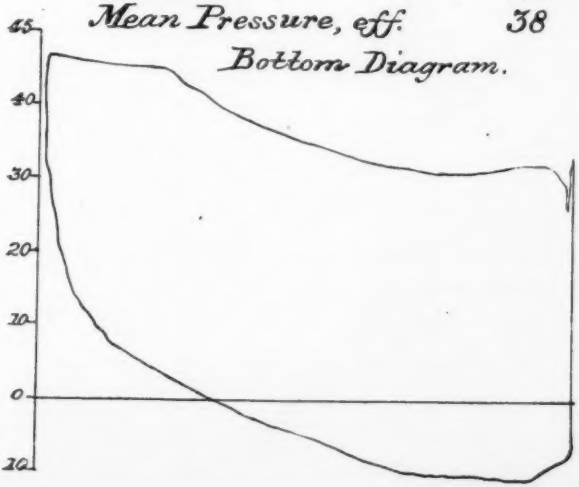
Vacuum 23



Top Diagram.

Vacuum 23

Mean Pressure, eff. 38

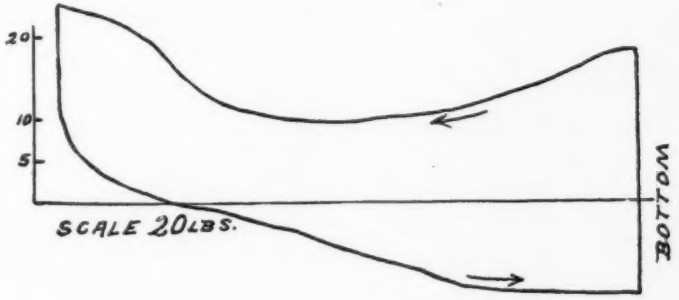


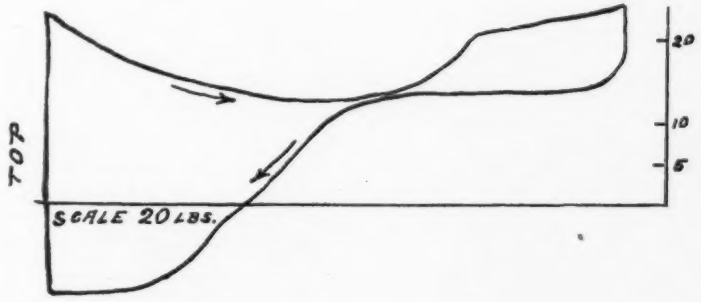
Bottom Diagram.

INDICATOR CARDS FROM BLAKE AIR PUMP.

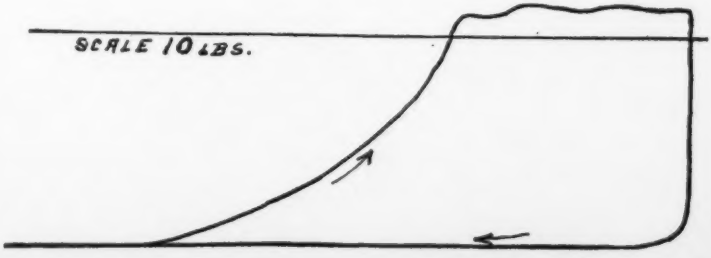
(CENTRE ENGINE ROOM.)

DOUBLE STROKES PER MIN.	$12\frac{1}{2}$
VACUUM IN INCHES	$25\frac{1}{2}$
I.H.P. OF BOTH STEAM CYLS.	16.28





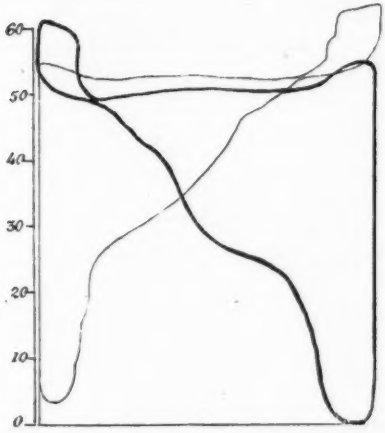
U. S. S. COLUMBIA.



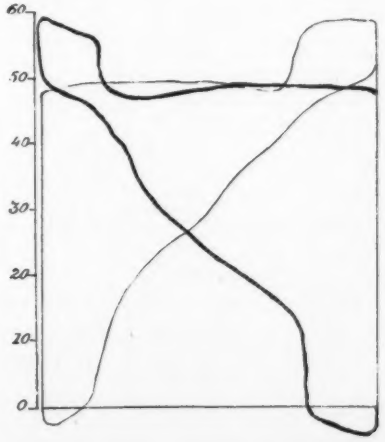
Nov. 18th '93 - 2.10 P.M.

U.S.S. Bancroft

Date and hour . . . Jan $\frac{26}{93}$ – 11.15 A.M.
Scale . . . 20
Double Stroke . . . 76
Mean Pressure, eff. . . . 20.34
Vacuum . . . . 26"



Date & hour . . . Jan. $\frac{26}{93}$ – 12.15 P.M.
Double Strokes 20
Mean Pressure, eff. 75
Vacuum . . . . . 25"

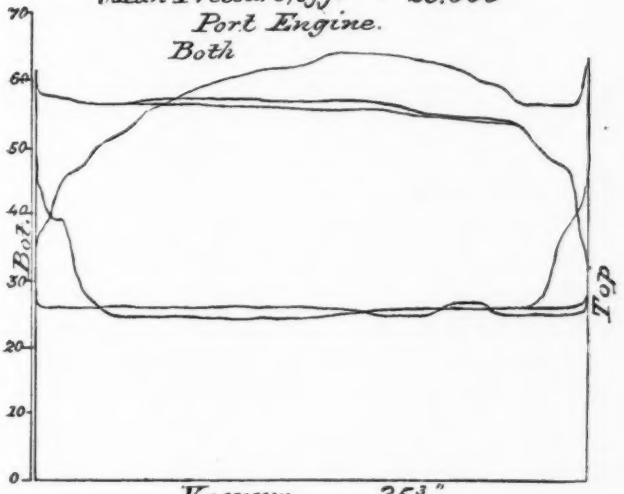


U.S.S. Castine.

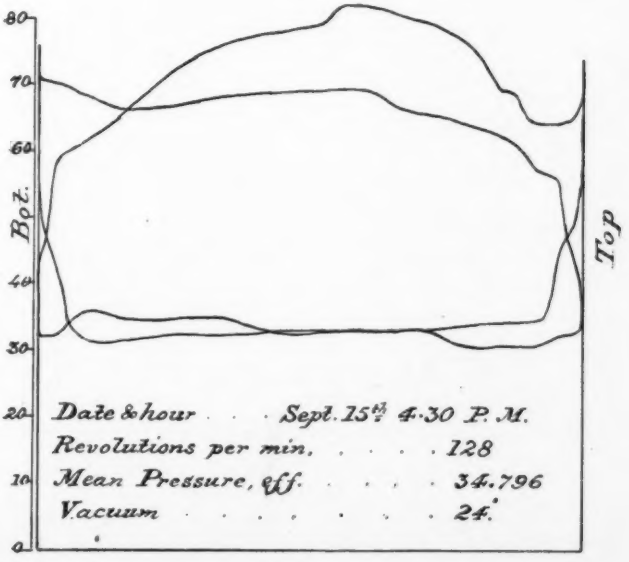
Date & hour . . Sept. 15, 2.30 P.M.
Revolutions per min. . . 104
Scale 20
Mean Pressure, eff. 29.666

Port Engine.

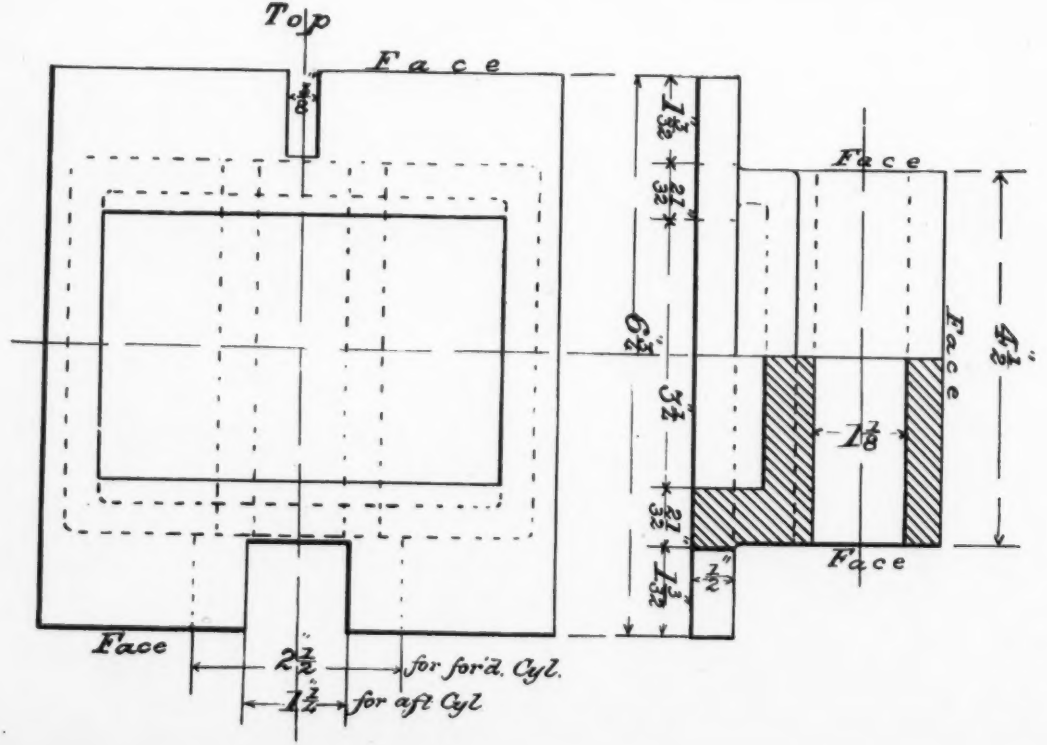
Both



Vacuum 25¾"



Date & hour . . Sept. 15th 4.30 P.M.
Revolutions per min. 128
Mean Pressure, eff. 34.796
Vacuum . . . . . 24.


Top
Face
1/8"
Face
Face
2 1/2" for forid. Cyl.
1 1/2" for aft Cyl
1 3/32"
21/32"
6 1/4"
3 1/4"
21/32"
1 3/32"
1/2"
Face
Face
Face
4 1/2"
1 7/8

MACHINERY DATA.

Name of Vessel.	Main Engines.			Air Pump Engines			A
	Type.	Number and diameter of cylinders.	Stroke.	Type.	Number and diameter of cylinders.	Stroke.	Type.
		ins.	*ins.*		*ins.*	*ins.*	
Atlanta	Hor. 3-cyl. Comp.	54, 74, 74	42	Horizontal B. A.	2 — 18	30½	Horizontal D. A. ind
Boston	Hor. 3-cyl. Comp.	54, 74, 74	42	Horizontal B. A.	2 — 18	30½	Horizontal D. A. ind
Chicago	Ver. Bm. Comp. T. S.	45, 78	57	Horizontal	1 — 22	...	Vertical S. A. combi
Charleston	Hor. Comp. T. S.	44, 85	36	Vertical Simple	2 — 10	16	Vertical S. A. ind.
Dolphin	Ver. Comp	42, 78	48	Horizontal Simple	1 — 18	12	Vertical S. A. combi
Petrel	Hor. Comp.	25, 46	33	Worked from main eng.	.....	...	Inc. S. A. attached t
Baltimore	Hor. 3-x T. S.	42, 60, 94	42	Worked from L. P. piston	.....	...	Hor. D. A. attached
Bennington	Hor. 3-x T. S.	22, 31, 50	30	Horizontal	1 — 14	12	Vertical S. A. combi
Concord	Hor. 3-x T. S.	22, 31, 50	30	Horizontal	1 — 14	12	Vertical S. A. combi
Newark	Hor. 3-x T. S.	34, 52, 76	40	Inclined Compound *	H.P. 11, L.P. 18	18	Vertical S. A. ind.
Philadelphia	Hor. 3-x T. S.	38, 58, 86	40	Inclined Compound *	H.P. 11, L.P. 18	18	Vertical S. A. ind
San Francisco	Hor. 3-x T. S.	42, 60, 94	36	Vertical Compound	H.P. 12, L.P. 19	18	Horizontal S. A. ind.
Yorktown	Hor. 3-x T. S.	22, 31, 50	30	Inclined Compound	H.P. 11, L.P. 18	12	Vertical S. A. combi
Bancroft	Ver. 3-x T. S.	13½, 21, 31	20	Vertical Simple	1 — 8	12	Vertical S. A. ind
Castine	Ver. 3-x T. S.	15¾, 22½, 35	24	Vertical Simple	2 — 6¼	12	Vertical S. A. ind.
Columbia	Ver. 3-x Triple Screw	42, 59, 92	42	Vertical Simple	2 — 16	21	Vertical S. A. ind.
Detroit †	Ver. 3-x T. S.	26½, 39, 63	26	Vertical Simple	2 — 7	15	Vertical S. A ind.
Machias	Ver. 3-x T. S.	15¾, 22½, 35	24	Vertical Simple	2 — 6¼	12	Vertical S. A. ind.
Marblehead	Ver. 3-x T. S.	26½, 39, 63	26	Vertical Compound	H.P. 6, L.P. 12	15	Vertical S. A. ind.
Minneapolis	Ver. 3-x Triple Screw	42, 59, 92	42	Vertical Simple	2 — 16	21	Vertical S. A. ind.
Monterey	Ver. 3-x T. S.	27, 41, 61	30	Vertical Simple	2 — 12	20	Vertical S. A. ind.
Montgomery	Ver. 3-x T. S.	26½, 39, 63	26	Vertical Simple	2 — 7	15	Vertical S. A. ind.
New York	Ver. 3-x T. S. Double	32, 47, 72	42	Vertical Simple	2 — 12	18	Vertical S. A. ind.
Olympia	Ver. 3-x T. S.	42, 59, 92	42	Vertical Simple	3 — 5½	20	Vertical S. A. ind.

* Bilge pumps attached to air pumps. † Auxiliary condenser not in use.

TR

Air Pumps.			Main Engines.												
	Number and diameter of cylinders.	Stroke.	Revolutions per minute.			Piston speed, feet per minute.			I. H. P.			Capacity of L.P. cylinders per minute.			
	ins.	*ins.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.* *cubic feet.*	*p.*	
A. ind.....	2 — 18	31¾		68.8			481.6			.....			28,770		
A. ind.....	2 — 18	31¾		72.2			505.4			.....			30,190		
combined.....	2 — 24½	21	69.95		70.75	664.4		672.0		.....		22,050		22,300	1
nd.....	2 — 24	16	115.4		113.9	692.10		683.70	3,159.5		3,156.5	27,270		26,940	1
combined.....	2 — 24½	21		74.15			593.2			2,144.2			19,686		
hed to eng...	1 — 14	18		126.78			824.0			1,000.21			9,509		
ched to eng..	1 — 28	42	116.4		116.18	814.8		812.7	4,845.59		4,985.48	39,270		39,170	2
ombined.....	2 — 18	12	150.9		151.1	754.5		755.5	1,642.4		1,680.4	10,288		10,300	
combined.....	2 — 18	12	151.5		153.3	757.5		766.5	1,678.1		1,636.2	10,328		10,450	
nd.....	2 — 24	12	127.3		126.7	848.8		844.7	4,468.4		4,113.9	26,740		26,610	2
nd	2 — 24	18	119.6		119 6	797.4		797.4	4,247.0		4,286.2	32,170		32,170	2
. ind.....	2 — 26	18	125.8		123.8	754.78		743.65	4,788.1		4,792.6	36,378		35,840	2
ombined.....	1 — 24	12	158.0		155.9	789.9		779.7	1,579.7		1,685.6	10,770		10,630	
nd	2 — 14	12	223.1		221.8	743.5		739.2	588.4		594.9	3,897		3,875	
nd.....	2 — 13½	12	217.0		220.6	868.0		882 4	1,076.4		1,051.8	5,800		5,896	
nd.....	2 — 31½	21	134.	127.7	132.9	938.	893.8	930.3	6,605.8	5,825.6	5,559.7	43,300	41,260	42,930	3
nd.....	2 — 14⅜	15	170.1		170.1	737.1		737.1	2,599.5		2,554.8	15,955		15,955	1
nd.....	2 — 12½	12	218.6		214.3	874.2		857.1	867.6		926.9	5,841		5,727	
nd.....	2 — 14⅜	15	179.0		173.4	775.6		751.3	2,435.6		2,427.8	16,787		16,262	1
nd.....	2 — 31½	21	131.9	131.1	132.2	923.6	917.7	925.3	6,586.7	7,218.8	6,560.7	42,637	42,358	42,377	3
ad.....	2 — 20	20	162.9		161.2	814.3		805.9	2,503.9		2,483.0	16,525		16,355	1
nd.....	2 — 14⅜	15	179.7		180.8	778.7		783.4	2,763.3		2,720.9	16,835		16,960	1
nd.....	2 — 25	18	{134.9 {134.9		135 0 135.0	942.6 942.6		945.0 945.0	f. 4,295.1 f. 4,340.6		a. 4,170.6 a. 4,141.0	f. 26,650 f. 26,650		a. 26,720 a. 26,720	f. 2 f. 2
nd.....	3 — 20	20	139.9		138.5	979.3		969.6	8,297.6		8,552.2	45,205		44,730	4

AIR PUMP DATA FROM RESULTS OF TRIALS OF U. S. VESSELS.

TRIAL DATA.

Main Engines.

I. H. P.				Capacity of L.P. cylinders per minute.			Water used per minute at 20 pounds per I. H. P. per hour, main engines, air and circulating pumps.			Double strokes per minute.			Piston speed, feet, per minute.		
p.	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	
		.....			*cubic feet.* 28,770			*cubic feet.* 14.686			.....			.....	
		.....			30,190			21.030			26.			137.6	
672.0		.....		22,050		22,300	13.16		11.44	44.5		43.6	155.7		
683.70	3,159.5		3.156.5	27,270		26,940	17.27		17.05	134.7		135.2	359.1		
		2,144.2			19,686			11.67			30.71			107.5	
		1,000.21			9,509			5.367			126 78			380.3	
812.7	4,845.59		4,985.48	39,270		39,170	26.00		26.75	116.42		116.08	814.8		
755.5	1,642.4		1,680.4	10,288		10,300	8.87		9.07	57.7		57.3	115.4		
766.5	1,678.1		1,636.2	10,328		10,450	9.04		8.81	51.4		50.9	102.8		
844 7	4,468.4		4,113.9	26,740		26,610	24 11		22.25	105.1		125.2	315.3		
797.4	4,247.0		4,286.2	32,170		32,170	23.01		23.21	105.3		118.2	315.9		
743.65	4,788.1		4,792.6	36,378		35,840	26.01		25.90	197.0		131.0	591.0		
779.7	1,579.7		1,625.6	10,770		10,630	8.79		8.99	123.0		120.5	246.0		
739.2	588.4		594.9	3,897		3,875		6.35			75.5			151.0	
882 4	1,076.4		1,051.8	5,800		5,896		11.36			120.5			241.0	
930.3	6,605.8	5,825.6	5,559.7	43,300	41,260	42,930	35.45	31.25	29.86	17.	14.4	13.9	59 5	50.4	
737.1	2,599.5		2,554.8	15,955		15,955	14.02		13.77	151.3		143 5	378.3		
857.1	867.6		926.9	5,841		5,727		9.76			223.5			447.0	
751.3	2,435.6		2,427.8	16,787		16,262	14.11		13 11	157.0		161.5	392.5		
925.3	6,586.7	7,218.8	6,560.7	42,637	42,358	42,377	35.28	38.67	35.12	15.	13.7	18.6	52.5	47.9	
805.9	2,503.9		2,483.0	16,525		16,355	13.59		13 50	85.6		84 7	285.3		
783.4	2,763.3		2,720.9	16,855		16,960	14.83		14.64	156.8		125.0	392.0		
945.0	f. 4,295.1		a. 4,170.6	f. 26,650		a. 26,720	f 23.07		a. 22.38	f. 19.5		a. 13.5	f. 58.5		a.
945.0	f. 4,340.6		a. 4,141.0	f. 26,650		a. 26,720	f. 23.29		a. 22 21	f. 17.6		a. 12.3	f. 52.8		a.
969.6	8,297.6		8,552.2	45,205		44,750	44.77		46.23	126.		124.7	420.0		

Air Pumps.																				
		I. H. P.			Capacity of pump cylinders per minute.			Vacuum in condenser.			I. H. P. of main engines and air and circulating pumps.			I. H. P. of air pumps + I. H. P. of main engines, air and circulating pumps.		
					cubic feet.			*ins.*											
s.	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*
.....			.....			.....			.....			2,749.			.....	
37.6			.....			243.2			.....			3,936.			.....	
	152.6		.....		509.7		499.6		.....		2,464.2		2,142.1		.....	
	360.5		.....		1,128.0		1132.5	26.2		26.1	3,233.3		3,219.2		.....	
7.5			41.2			351.9			25 5			2,185.4			.01885	
0.3			.....			203.3			.....			1,004.48			.....	
	812.7		.....		1,742.0		1,737.5	24.5		24.3	4,868.80		5,008.69		.....	
	114.6	19.14		17.48	203.9		202.5	23.98		23.64	1,661.6		1,697.9	.01153		01030
	101.8	27.33		.....	181.6		179.9	24 11		24.00	1,691.8		1,649.8	.00825		.....
	375.6	29.97		31.60	990.5		1,180.0	25.97		25.66	4,513.4		4,160.6	.00664		.00760
	354.6	31.56		33.32	992.4		1,114.0	24.3		24.7	4,308.8		4,344.2	.00733		.00767
	393.0	66.48		41 23	2,179.0		1,449.0	25.7		26.1	4,868.9		4,848.9	.01365		.00851
	241.0	60.88		57.91	386.4		378.5	24.8		25 0	1,640 6		1,683.5	.03712		.03441
1.0			5.09			161.4			26 3			1,189.8			.009558	
1.0			14.07			239 5			25.3			2,162.			.013018	
0.4	48.6	22.17	17.97	16.30	322 0	272.7	263 0	25.1	25.0	25.6	6,636.7	5,850.5	5,590.0	.00334	.00307	.00292
	358.7	19.83		18.94	426.3		404.2	25.4		25.1	2,624.63		2,577.90	.00756		.00735
7.0			17.94			444.3			25 7			1,828.2			.00981	
	403.7	15 65		15.95	442.3		459.0	26.2		25.9	2,641.1		2,454.2	.00593		.00650
7.9	65.1	10 91	8.39	13.87	284.1	259.4	351.9	24.7	25.3	25.1	6,615.6	7,251.1	6,586.6	.00165	.00191	.00127
	282.3	18.37		18.16	622.3		615 8	26.6		26.2	2,544.6		2,527.1	.00722		.00719
	312.5	18.82		15.00	441.8		352.2	25.75		26.10	2,776.7		2,740.2	.00678		.00548
	a. 40 5 a. 36.9	f. 13.07 f. 9.51		a. 9 19 a 6 97	f 199 4 f 179.9		a. 138.3 a. 125.8	f 25.00 f. 25 20		a. 25 60 a. 25.80	f. 4,318.0 f. 4,360.0		a. 4,189.7 a. 4,157.8	f. .00303 f. 00218		a. .00219 a. .00168
	415.6	18.22		35 78	1,374.0		1359.8	24.94		25.59	8,382.3		6,653.2	.00217		.00414

Air Pumps. Double strokes per minute. c.	p.	Piston speed, feet, per minute. s.	c.	p.	I. H. P. s.	c.	p.	Capacity of pump cylinders per minute. cubic feet. s.	c.	p.	Vacuum in condenser. ins. s.	c.	p.	I. H. P. of main engines and air and circulating pumps. s.	c.	
......			......			......			......			......			2,749.	
26.			137.6			......			243 2			......			3,936.	
	43.6	155.7		152.6		......		509.7		499.6		......		2,464.2		2,1
	135.2	359.1		360.5		......		1,128.0		1132.5	26.2		26.1	3,233.3		3,2
30.71			107.5			41.2			351.9			25 5			2,185.4	
126 78			380.3			......			203.3			......			1,004.48	
	116.08	814.8		812.7		......		1,742.0		1,737.5	24.5		24.3	4,868.80		5,0
	57.3	115.4		114.6	19.14		17.48	203.9		202.5	23.98		23.64	1,661.6		1,6
	50.9	102.8		101.8	27.33		......	181.6		179.9	24 11		24.00	1,691.8		1,6
	125.2	315.3		375.6	29.97		31.60	990.5		1,180.0	25.97		25.66	4,513.4		4,1
	118.2	315.9		354.6	31.56		33.32	992.4		1,114.0	24.3		24.7	4,308.8		4,3
	131.0	591.0		393.0	66.48		41 23	2,179.0		1,449.0	25.7		26.1	4,868.9		4,8
	120.5	246.0		241.0	60.88		57.91	386.4		378.5	24.8		25 0	1,640 6		1,6
75.5			151.0			5.09			161.4			26 3			1,189.8	
120.5			241.0			14.07			239 5			25.3			2,162.	
14.4	13.9	59 5	50.4	48.6	22.17	17.97	16.30	322 0	272.7	263 0	25.1	25.0	25.6	6,636.7	5,850.5	5,5
	143 5	378.3		358.7	19.83		18.94	426.3		404.2	25.4		25.1	2,624.63		2,5
223.5			447.0			17.94			444.3			25 7			1,828.2	
	161.5	392.5		403.7	15 65		15.95	442.3		459.0	26.2		25.9	2,641.1		2,4
13.7	18.6	52.5	47.9	65.1	10 91	8.39	13.87	284.1	259.4	351.9	24.7	25.3	25.1	6,615.6	7,251.1	6,5
	84 7	285.3		282.3	18.37		18.16	622.3		615 8	26.6		26.2	2,544.6		2,5
	125.0	392.0		312.5	18.82		15.02	441.8		352.2	25.75		26.10	2,776.7		2,7
	a. 13.5 a. 12.3	f. 58.5 f. 52.8		a. 40 5 a. 36.9	f. 13.07 f. 9.51		a. 9 19 a 6 97	f 199 4 f 179.9		a. 138.3 a. 125.8	f 25.00 f. 25 20		a. 25 60 a. 25.80	f. 4,318.0 f. 4,360.0		a. 4,1 a. 4,1
	124.7	420.0		415.6	18.22		35 78	1,374.0		1359.8	24.94		25.59	8,382.3		8,6

RESULTS.

	I. H. P. of air pumps ÷ I. H. P. of main engines, air and circulating pumps.			Capacity of air pumps, per minute ÷ capacity of L.P. cylinder per minute.			Cubic feet of water used per minute at 20 pounds per I.H.P. ÷ air pump capacity per minute.			Capacity of air pumps per minute ÷ I.H.P. of main engines, air and circulating pumps.		
p.	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*	*s.*	*c.*	*p.*
		.....			.....			.....			.....	
		.....			.008056			.08636			.0618	
2,142.1		.....		.02312		.022405	.02583		.0229	.2069		.2332
3,219.2		.....		.04137		.04204	.01531		.01506	.3488		.3518
		.01885			.01788			.03318			.16102	
8		.....			.02138			.0264			.2025	
5,008.69		.....		.04437		.04437	.01493		.01540	.3579		.3470
1,697 9	.01153		.01030	.01982		.01966	.0435		.04479	.1227		.1193
1,649.8	.00825		.....	.01758		.01722	.04978		.04898	.10735		.10903
4,160 6	.00664		.00760	.03705		.04434	.02434		.01886	.21945		.2836
4,344.2	.00733		.00767	.03085		.03463	.02622		.02083	.2303		.2565
4,848.9	.01365		.00851	.05990		.04043	.01194		.01787	.4476		.2989
1,683.5	.03712		.03441	.03588		.03561	.02275		.02375	.2355		.2248
		.009558			.04154			.03935			.13565	
		.013018			.04096			.04744			.1108	
5,590.0	.00334	.00307	.00292	.007437	.006618	.006127	.1101	.1146	.1135	.04856	.04661	.04705
2,577.90	.00756		.00735	.02672		.02533	.0329		.0341	.16240		.1568
		.00981			.03841			.02197			.24305	
2,454.2	.00593		.00650	.02635		.02823	.0319		.02855	.1675		1871
6,586.6	.00165	.00191	.00127	.00666	.00612	.00830	.1242	.1491	.0998	.04294	.03573	.05342
2,527.1	.00722		.00719	.03766		.03761	.02184		.02193	.276		.2437
2,740.2	.00678		.00548	.02621		.02077	.03357		.04157	.1591		.1285
a. 4,189.7	f. .00303		a. .00219	f. .00748		a. .00518	f. .1157		a. .1618	f. .0462		a. .0330
a. 4,157.8	f. 00218		a. .00168	f. 00675		a. .00471	f. .1295		a. .1766	f. .0413		a. 0303
8,653.2	.00217		.00414	.0304		.03039	.03257		.0340	.1639		.1572

were at right angles. When the cranks were nearly opposite this was not necessary, as one or the other of the pistons was always on the working stroke. These valves have answered the purpose for which they were designed, and the pumps work fairly well, but of course better results could be obtained with the wire-drawing in the port itself, so that both the steam and exhaust would be wire-drawn. The cards shown for the *Castine* were taken after the steam cylinders of her air pumps had been fitted with similar valves. This kind of a valve would have been unnecessary if the ports had been made from $\frac{1}{40}$ to $\frac{1}{50}$ of the area of the cylinder instead of $\frac{1}{10}$, so that there would have been considerable wire-drawing of the steam. It makes no particular difference whether you do all the wire-drawing in the steam port, or in the exhaust, or part in one and part in the other.

Should you have occasion to construct an independent fly-wheel pump use two cylinders, put the cranks nearly opposite, although right angles will work fairly well. Use a fly wheel as heavy as the *Charleston*, in proportion, if circumstances will permit. Set the valves to follow full stroke, and be sure not to have any compression. Make the port area from $\frac{1}{40}$ to $\frac{1}{50}$ of the area of the cylinder, and either put cocks or valves in the ports, so that they can be varied in size until the most suitable area has been found for the particular speed at which the pump is to run, or make the main valves so that their port areas can be regulated. Make as little clearance between the throttle valve and the main valves as possible. You will then have a pump that will run smoothly and without danger of stopping.

Reliability.—By this we mean the freedom from derangement, and general certainty of action. It is annoying, to say the least, to have a pump stop unless some one shuts the steam off. We don't want to have to detail some man to stand by the air-pump throttle whenever we are working the engine to signal, or at any other critical time. This quality is of far more importance than either weight or economy, and whenever there is a marked difference between the reliability of two pumps the more reliable should always be taken. We have both fly-wheel and direct-acting pumps in our service. We have had trouble with individual

pumps of each type, but I think it could always be traced to faults in the design or workmanship of the individual pump. Since the fly-wheel pump has, with an equal amount of wire-drawing, a much steadier motion, as a glance at those cards will show, there is no doubt that the fly-wheel pumps can be run at a much higher speed with safety than the direct-acting ones. In direct-acting pumps there are little ports and valves which let the steam act upon a small steam piston which moves the valve. If well cared for, this usually gives no trouble, but sometimes these little ports get stopped up, or the valve seat gets corroded away, or the little piston gets rusted fast so it will not move. These are avoided in the fly-wheel pumps, since the valves are moved by an eccentric and have to go if the pump does; but as an offset we have crank pins, crank shafts, and crosshead journals which may, and sometimes do, heat and give trouble.

With equal care, and a reasonably good design, I see no reason why one class of pump should be more reliable, or less liable to get out of order than the other.

Annexed is a table of some of the principal data of our air pumps on the trial trips.

Discussion.

Chief Engineer McKean, U. S. Navy.—The thoughtful and able paper by Mr. Bailey seems to leave little to be said in the way of discussion. I would state, however, that the theoretically excellent suggestion to connect the clearance spaces at each end of the pump with the barrel of the pump by means of ports cut through the liner, and which ports are to be overrun by the piston at each end of the stroke, may involve care in keeping those ports clear, if, as I presume, they are small, and therefore liable to become choked by material from the packing of the piston, from the products of lubrication in the main cylinders, or by dirt from other sources; still, such clearing would only form one more item to attend to in the routine examinations of piston, valves, etc. Mr. Bailey does not consider the friction of piston packing, and only alludes in passing, to that through the valves;

and he is right, from his point of view, in ignoring those frictions, since they exist about equally in all ordinary pumps. Some years ago I studied upon a pump-design intended to modify these sources of loss and trouble, by doing away with the piston and valves as commonly fitted; but the improvements made upon the centrifugal variety seemed to include the advantages aimed at, at least for most purposes. The plan was to have a sheet-rubber diaphragm across a pump chamber of any shape of section, at its middle point and athwartships, and to attach the pump rod to the center of the diaphragm. If the stroke was kept short, as is generally the tendency in auxiliary pumps, the flexibility of the rubber might be sufficient; if not, or if a longer stroke was required, an accordion action might be secured by concentric folds in the diaphragm. In place of the ordinary valves there were to be thin rubber tubes of oblong section, so that the water would be drawn through the receiving tubes and forced through the delivery ones by the motion of the diaghragm in one direction, while the pressure due to the reverse motion would force the lips together and prevent the return of the water. Leakage at the stuffing box was to be lessened by a sheet-rubber sleeve of sufficient length to accommodate itself to the stroke, secured to pump bonnet at one end and to pump rod at the other, and enclosing the ordinary stuffing box. But to return to the paper under discussion: we owe Mr. Bailey many thanks for his long and interesting contribution towards a subject which has become one of the most important in marine engineering.

Chas. H. Haswell, Esq.—It would afford me much pleasure to express my approval of the views of the author of this valuable contribution, both to the physics and mechanics of the steam engine, in a more extended manner than my engagements will admit of, as I am not only about leaving the city for an uncertain period, but will be employed in a manner that precludes consideration of the subject. I have to restrict myself, therefore, to a brief expression regarding the air pump, and the mechanism of its construction.

The air pump of a condensing and marine steam engine,

whenever practicable, should be operated independently of it, not only to meet various conditions of the engine, but to be available as an exhausting pump to free a vessel from water, and also as a fire pump.

Regarding the design of a pump, the requirements of a normal condition of construction and operation are a single cylinder set vertical, with or without a fly wheel, according to its design and construction, proper proportion of diameter to stroke, high finish of its frictional parts, with the least practicable clearance space, as low speed as practicable, and with water valves of hard rubber, or faced with it—fly and poppet valves of metal will sand, and slide will cut. Duplicate or triplicate cylinders involve additional cost of construction, operation and space.

The design of the gun-boat pump, to my limited knowledge of what has been attained of late years, is exceptionally good. It should be very firmly seated and on a broader basis.

Passed Assistant Engineer W. M. McFarland, U. S. Navy.—I have read my friend Bailey's article with great pleasure and satisfaction, and I have no hesitation in saying that in my judgment it is the most useful and instructive paper on the subject that has appeared in a long time. He treats the matter so clearly and completely that really it only remains to give confirmatory details. I am somewhat familiar with the struggles to make the crank air pump satisfactory, and, so far as I know, until he called attention to the point no one had thought of the source of trouble being in the valve gear. His article, however, explains the whole matter so clearly that we can all see now where the trouble lay.

Recently I have had an opportunity to study the action of a crank air pump whose working is far from satisfactory, and a few details with regard to it may be of interest. The pump has two horizontal water cylinders, single-acting, driven by a vertical compound engine, with cranks at right angles, steam valves of usual proportions and driven in the usual way. Attached to the air pump piston rods are bilge pumps, also single acting. I understand that the original design called for double-acting pumps,

and I believe if they had been fitted the machine would have worked much more smoothly, for there would then have been four points of maximum work per revolution, symmetrically distributed, while now there are but two, and these come in the same half revolution, accentuated by the single-acting bilge pumps whose discharge strokes coincide with those of the air pumps. The fly wheels are very small. As a result of the combination of all these circumstances, it is necessary to run the pumps at from 75 to 100 revolutions per minute to keep them from stopping, while any change in the working of the main engines will cause them to stop unless the throttle is properly adjusted. In fact, I am told that a man always remains near them to open the throttle in time.

Now, it has struck me in looking at this case that the arrangement of cranks is wrong, caused, doubtless, by failing to recognize the peculiarities of the case. Where a uniform turning effort is desired to do perfectly uniform work, the arrangement of cranks at right angles is the best; but in driving a couple of single-acting air pumps the work is not only not uniform, but extremely unequal. On the discharge stroke we have a good deal of work towards the end, and very little at the beginning, while on the other stroke we are simply passing the water through the bucket valves and practically doing nothing. Now, when the cranks are opposite, or nearly so, we have some work on each engine stroke, and have both steam cylinders to do it, while with cranks at right angles all the work is crowded into less than half a revolution, and for the other half the steam pistons simply run away, unless there is a very heavy fly wheel. It seems to me that this has something to do with the admirable working of the Blake air pumps on the *New York*, *Columbia* and *Minneapolis*. There the pistons are directly opposite, and being connected by a beam, we get the full effect of both steam pistons for the working stroke of each pump.

A good deal has been said about the lack of economy of steam engines working as slowly as the ones of these pumps do, and doubtless they do use a good deal of steam per I.H.P., but their I.H.P. is so small compared with that of most crank pumps

that there can be no doubt that *the steam used per hour* is much less than in the crank pumps.

The pumps I have first mentioned are almost exactly the same size as those of the *New York*, and it is very interesting to compare them.

	Crank Pump.	Blake Pump.
Steam cylinders, number and diameter, inches.	Compd. 12 and 19	2 of 12
Water cylinders, number and diameter, inches	2 of 26	2 of 25
Stroke, inches	18	18
I.H.P. of main engines to which connected	4,800	4,200
Double strokes of pumps per minute	164	15
I.H.P. of pumps (steam cylinders)	54	95

I am sure we shall all agree with Mr. Bailey when he says regarding reliability: "This quality is of far more importance than either weight or economy, and whenever there is a marked difference between the reliability of two pumps, the more reliable should always be taken." This would hold true for any ship, but it is specially true of war vessels. Squadron steaming in time of peace makes the lives of the engine room force weary enough from the constant changes required to keep position, without the added worry of having an air pump that may stop, while the conditions of actual combat require still more that the air pump should run itself with practically no attention. I feel that the Society is to be congratulated on this able paper, and I hope its discussion will be in keeping with the merit of the paper.

Robert Forsyth, Esq., Associate.—I consider the paper on air pumps by Mr. Bailey a most valuable one to marine engineers, chiefly because it brings prominently to notice the many difficulties connected with air pumps. Much misdirected ingenuity has been expended upon the effort to design an air pump of light weight and occupying small space, which could be relied upon to work efficiently at different speeds and loads, and which would require the minimum of attention.

The necessary capacity of an air pump is, as Mr. Bailey says, a very uncertain element; but, in air-pump design, increase of capacity cannot possibly make up for *lack of efficiency*.

The vertical single-acting pump attains, on the whole, the highest degree of efficiency with the minimum of displacement. It can be designed to be easily accessible for repairs, and there are few wearing parts.

When rubber valves are used, I prefer to use them without springs, as I have found that springs cause more wear on the back of the valve than the seating causes on the surface; and broken springs are a fruitful source of trouble with valves, guards and studs. By reducing the lift of a 4½-inch rubber valve to ¼ inch, and making the guard flat, with at least as much surface as the valve seat, the rubber valve will last indefinitely, unless the pump is driven from the main engine, in which case "racing" changes the conditions very materially.

Professor Hollis' idea of making a by-pass on the barrel of the double-acting air pump will work very satisfactorily. A somewhat similar arrangement has been used by Buckhardt and Weiss on some of the sugar plantations of the Hawaiian Islands, where large double-acting pumps were improved by adding a slide valve operated by cam movements to control a communication to opposite ends of the bucket during the first inch of the return stroke of the bucket. Professor Hollis' arrangement will accomplish the same result in a simple manner, with no wearing parts to get out of order.

While the operation of the air pump from the main engine is, for the designer, the simplest, lightest, and most economical, there are many practical objections to it, and many good reasons for using air pumps independently moved.

Taking the uses of the air pump preparatory to and during a long voyage in their order—

1st. In getting up steam and warming the main engines, the independent pump cares for the condensed steam without waste.

2d. In getting under way, and in promptly executing orders from the bridge, the main engines can be handled with a facility

which is impossible when the main engines have the air pump connected as part of itself.

3d. During the voyage, the independent pump need only be driven at a speed that will make a good vacuum; and not being subject to the tear and wear arising from the "racing" of the main engines, the valves and guards will be maintained in a reliable condition for a long period.

4th. When nearing port, and orders to stop are liable to come unexpectedly, the independent pump takes care of the surplus steam, which is led into the condenser instead of roaring off from the safety valves. The saving in fresh water thus effected, and also in aiding to care for the boilers, amply repays any small additional expense that may be charged to the operation of the independent pump.

Of the two systems of operating the air pump, the direct-acting has the advantages of simplicity and the possibility of running at very low speeds without risk of stopping at the end of the stroke.

The crank and fly-wheel engine must have a fly wheel of diameter and weight great enough to pass the centers without hesitation if the main-engine speed happens to be increased somewhat. There are some ships running out of San Francisco with air pumps driven by independent compound engines which work satisfactorily at sea, but they are fitted with 10-foot fly wheels weighing six tons, while the main engines only indicate from 2,000 to 3,000 horse-power. The additional labor to care for the journals, the weight and the space occupied by the fly wheel, are serious objections to this design. With the direct-acting system, careful attention must be given to get nearly uniform length of stroke for the different speeds without adjustment of the tappets, and to attain this the diameter of the steam cylinder must be large enough to allow the requisite cushioning to occur during only a very short part of the stroke. This size will be found to be largely in excess of what is required to do the work of the pump. On the *Monterey's* pump, an actual test, from four strokes per minute to one hundred per minute, showed the variation per length of stroke to be $\frac{3}{32}$ of an inch.

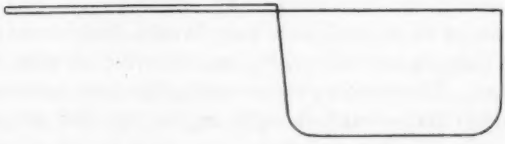
In the case of a vertical direct acting pair of air pumps recently erected by the Union Iron Works, the exhaust steam from the air pump steam cylinder was led by suitable nozzles into direct contact with the water in the hot-well discharge pipe, raising the temperature of the feed water from 112 to 170 degrees. This arrangement is quite economical in the consumption of steam for the amount of useful work done, and removes the objections often urged that the direct-acting pump is too extravagant in the use of steam for continuous work.

In either case, whether driven by a crank engine or direct, a good governor should be provided which can be readily adjusted to any required change of speed.

Professor Ira N. Hollis.—The subject of air pumps is a very interesting one, involving, as it does, some of the most difficult and troublesome questions in design, and I am glad to see Mr. Bailey take it up. The first point I notice, however, is in reference to the *Charleston's* pumps, and I must disclaim the merit of taking them in hand any more than they were taken in hand by the other engineers of the ship. We all discussed the merits of the pumps and the best methods of curing the tendency to stop. A good governor would have gone far towards the cure. It may be as well to state briefly the history of these pumps, as they were designed at the Union Iron Works after several other pumps had proved failures. The design, purchased in England by the Government, was exceedingly unsatisfactory, and, as supplied to the Union Iron Works, little could be made of it. The engines were compound, horizontal, with Marshall valve gear. The steam cylinders were placed between the crank-shaft, which had cranks at right angles, and the air and bilge pumps. One double-acting air pump was connected with the high-pressure piston, and two plunger bilge pumps with the low-pressure piston. The whole was placed directly under the condenser. It was so evident that the pump was too small that the contractors increased it considerably in size; but even then, the result of the dock trial fell far below the government requirements. The vertical single-acting pump with two steam and

pump cylinders and a heavy fly wheel was afterwards put in by the Union Iron Works as a substitute for the English design. It worked admirably at the full power of the main engine, but had the fault of stopping without warning at low powers. The fly wheel was of great benefit, but as the pump was put in after a dock trial, it had to be placed at the end of the condenser with long channelways running athwartship to the suction. It worked at a great disadvantage when the ship was rolling, and the water reached it intermittently. When the ship rolled slowly, the pistons would pound heavily on the up stroke until relieved of the gulps of water which came from the condenser. The above does not seem to me a real defect in the pump, but in the conditions under which it worked. Another thing which seriously affected its working at low powers was the small throttle opening required for steam. A sudden demand could not be supplied, and the pump sometimes stopped before the throttle could be opened, another unfortunate condition of working. In fact, all conditions are unfortunate for machinery which is run at one-eighth the power for which it was designed. The success, economically and mechanically, of any machine lies mainly in the regularity of its running at the designed power. An air pump has to work under peculiarly hard conditions even in the

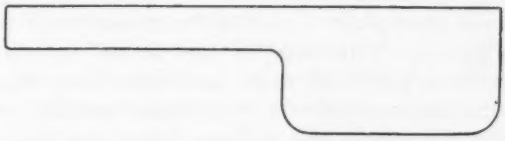
Fig. 1.



best of circumstances, both on account of the irregularity of its load and the mixture of air, vapor and water which passes through it. The pressures come just at the wrong time for an independent pump, and the fact that the cards from the steam and water cylinders are the reverse of each other makes the pump expensive in steam.

Some saving might be effected by designing the valve to remain open to steam part of the return stroke, so that a portion of the steam could be driven back into the boiler instead of the condenser. The steam valve could be closed at one-third or one-half stroke, and the exhaust valve opened, thus giving the card shown in Fig. 1. As this could be worked with only two steam cylinders and cranks opposite, a better arrangement could be devised by having the steam exhaust into the high-pressure receiver half the stroke and into the condenser the other half, giving a card like Fig. 2. The latter method could

Fig. 2.



be applied to a single cylinder, either single or double acting, and the effective steam pressures would be found approximately where they were wanted. A vertical single-acting pump does not usually require steam above the piston, as the weight of the moving parts will cause the down stroke. The cap brasses of the *Charleston's* pumps were never touched by the journals.

Mr. Bailey very properly places special stress upon the tightness of all joints and glands. As stated in his paper, the low pressure of steam in the intermediate and low-pressure cylinders, at the ordinary powers of naval vessels, permit inward leaks which do not exist at high powers. I have seen the vacuum improve with increase of power without any change whatever in the speed of the air pump.

The pressure required to lift foot valves points to mechanical means of opening these valves, or to ports to be uncovered by the piston for a certain part of the stroke, as in the pump without valves.

The independent pump in use at present seems much too large for the work, and a successful high-speed pump would

show gain in economy and great reduction in weight. The problem is complicated by the difficulty of getting water and even vapor under low pressure to follow a rapidly-moving piston, and by the churning which takes place in the pump barrel and clearances. The vaporous mixture left in the clearances of a high-speed pump is fatal to the formation of a good vacuum. A little calculation will show about the limit of speed before churning begins in a vertical pump with shaft and fly wheel. The piston reaches its maximum velocity near mid stroke, and is then constrained by external connections to slow down, but the water carried up by it slows down only by the action of gravity, and, therefore, may leave the piston (or be thrown upward by it) and reach the delivery valve ahead of it. This lifting of water takes place as soon as the retardation of the piston exceeds gravity. If the water strikes the delivery valves with sufficient force it will lift them and escape into the air, thus leaving the clearance full of air and vapor instead of water, and effectually destroying the vacuum above the piston. The velocity of the pump would have to be excessive to cause a water pressure equal to the atmosphere in this way; but, as a matter of fact, at the usual velocities, the vapor within the pump has already reached the atmospheric pressure before the water leaves the piston and strikes the valves above. In such a case the valves will open with a very slight water pressure and let the water escape. I have seen cases of heavy pounding on the upstroke of the piston, as no air or vapor escaped with the water to cushion against the pump bonnet. The pounding was relieved at once by admitting a small amount of air into the pump through the indicator cock.

The point of the stroke where this lift of water takes place may be determined as follows:

Let N = number of revolutions per minute,

r = length of crank in feet,

s = distance of the piston from the end of its stroke for the crank angle θ,

$s = r - r \cos \theta$, neglecting the connecting rod,

$\frac{ds}{dt} = r \sin \theta$ = velocity of piston,

$\frac{d^2s}{dt^2} = r \cos\theta \left(\frac{d\theta}{dt}\right)^2$ = acceleration or retardation of piston,

$\frac{d\theta}{dt}$ = angular velocity of crank $= \frac{2\pi N}{60} = \frac{\pi N}{30}$;

Therefore, when

$$g = \frac{d^2s}{dt^2} = r \cos\theta \left(\frac{\pi N}{30}\right)^2, \qquad (1)$$

we have reached the limit where the piston begins to throw the water upwards. The exact position of the piston, or the crank angle, can be determined from this equation if r and N be given, or the limiting number of revolutions for highest efficiency can be found if r and $\cos\theta$ be given.

Take the case of the *Charleston's* pumps: $r = \frac{5}{8}$ foot, and the piston is about one-eighth of the stroke from the end when the vapor and air reach atmospheric pressure.

Then $$s = r - r\cos\theta = \frac{\text{stroke}}{8} = \frac{r}{4},$$

and $$\cos\theta = \tfrac{3}{4}.$$

Substituting in equation (1), we get

$$g = \frac{3}{4}r \cdot \frac{\pi^2 N^2}{900} = \frac{\pi^2 N^2 r}{1200},$$

$g = 32.2$, and $r = \frac{5}{8}$.

$$32.2 = \frac{5\pi^2 N^2}{8 \times 1200}, \quad N^2 = \frac{32.2 \times 8 \times 1200}{5\pi^2};$$

or $$N = 79 \text{ (about)}.$$

Thus the greatest efficiency of the pump would be reached at 79 revolutions for the low powers of the main engine. As a matter of fact, this calculation does not take into consideration all of the circumstances; and the case is more favorable for high speed in a single acting pump, as the delivery chamber is separate from the suction chamber, and a bad vacuum above the piston does not cause a bad vacuum below. An instance is

given before where the indicator cock above the piston was opened for air cushion, and the vacuum in the condenser was not impaired in the slightest degree. It was found entirely possible to run the pump at 100 revolutions without decreasing the vacuum.

The case against the double-acting horizontal pump is serious, however, as the water begins to leave the piston at mid-stroke and forms in a wave against the pump bonnet in readiness to escape as soon as the vapor behind it reaches atmospheric pressure. The clearance is thus left more or less filled with air and vapor depending upon the speed of the piston; and since the delivery chamber becomes the suction chamber on the return stroke, the vacuum falls. To avoid churning, therefore, the above shows the necessity of low speeds for horizontal pumps, and moderate speeds for vertical lifting pumps. Any method of getting the vapor out of the clearances will make a high-speed pump entirely possible.

The common reciprocating pump without fly wheel or shaft is better off in this matter of churning, as the piston has a fairly steady velocity from one end of the stroke to the other. It is very wasteful of steam and usually much too large.

Fred'k Meriam Wheeler, Esq., Associate.—Mr. Bailey's paper is certainly an admirable one, and is especially interesting to me, as I have had some experience in the matter of independent air pumps; and it will give me great pleasure to discuss the subject at length at another time. I would like now to make reference to the indicator cards which Mr. Bailey gives from the steam cylinders of the Blake vertical twin air pumps on the cruiser *New York*. The cards that I have reference to were taken from the top ends of the steam cylinders. These cards, both in the steam line and the compression line, are rather unusual. The explanation of this is the fact that the cushion valves were not properly adjusted before the trial trip took place. The result was that there was entirely too much back pressure, and consequently the pumps required more steam pressure than was necessary to run them. On the trial trips of the *Columbia* and

Minneapolis, the cushion valves on the Blake air pumps were correctly adjusted, consequently the pumps showed much better cards, and took considerably less steam pressure to run them. I simply mention this matter in explanation of the peculiar appearance of the cards referred to.

George W. Dickey, Esq.—I have read this very excellent paper with a great deal of interest, the subject being one that has been repeatedly forced upon my attention under circumstances that presented no means of escape from the responsibility of deciding what next to do when an air pump could neither be coaxed nor forced into doing what it had been expressly built to do.

I agree entirely with the author as to what the functions of an air pump as an appendage of the steam engine are, and how such a pump should be constructed to perform those functions with the least expenditure of power. I am somewhat disappointed, however, that Mr. Bailey should have confined his paper to a comparison between the merits of direct-driven and crank-driven air pumps, as I think the main question with engine builders and designers at present is, should the air pump derive its motion from that of the main engine, or be an independent machine. It is needless for me to give my views on this question, as I could add nothing to the statement made in my paper read before the Engineering Congress at Chicago last year.

In regard to the comparative space occupied and weight of direct-acting and fly-wheel air pumps, Mr. Bailey says that the direct acting will probably represent an average saving of 30 per cent. in space, and from 30 to 40 per cent. in weight. I find that weight and space is very largely a question of design, and so far as my experience goes, the pump operated through a crank shaft need not weigh more than the pump operated direct. The weight of one set of air pumps for the *Charleston* complete as they left the shop was 15,795 pounds. The pumps are 24 inches diameter and 16 inches stroke. The volume displaced by one revolution was therefore 8.3 cubic feet; or the weight was 1,903 pounds per cubic foot displacement, and 4.78 pounds per indicated horse power of the main engines.

The weight of one set of air pumps of the Dow direct-acting type for the *Monterey* complete as they left the shop was 9,060 pounds. The pumps are 20 inches diameter and 20 inches stroke. The volume displaced by a double stroke or one revolution was therefore 7.2 cubic feet, making the weight 1,400 pounds per cubic foot displacement, and 3.55 pounds per indicated horse power of the main engines.

The weight of one set of air pumps—three pumps operated through a three throw crank shaft—for the *Olympia* complete as they left the shop was 13,480 pounds. The pumps are 20 inches diameter and 20 inches stroke. The volume displaced in one revolution is 10.8 cubic feet, equivalent to 1,250 pounds per cubic foot displacement, and 1.55 pounds per indicated horse power of the main engines; so that, while the crank shaft air pumps of the *Charleston* are heavier than the direct-driven air pumps of the *Monterey*, the crank-shaft air pumps of the *Olympia* are much lighter than either.

In regard to the space occupied, the crank and fly-wheel pumps, if the dimensions of the rectangular space is to be measured over the wheel, will naturally exceed that of the direct-driven pump; but it often occurs that floor space is of more importance than the cubic space occupied. This is the case with the air pumps of the *Charleston*, there being plenty of room above where the fly wheel is, but nothing to spare at the floor line.

In the table of comparisons on the opposite page, the cubic space occupied is the volume of the rectangular figure enclosing the largest dimensions, and from the examples there given it will be seen that the air pumps operated through a crank shaft need not occupy more space than those operated direct.

In regard to economy of working, I fully expected that Mr. Bailey could have compared the amount of steam used say by the *Monterey's* direct pumps with the *Olympia's* crank pumps, for which he had all the data at hand in the Bureau of Steam Engineering. I am not in this discussion advocating crank pumps, as I see no necessity for independent air pumps, but if they are to be used, let us have the best in efficiency and economy.

	Charleston.	*Monterey.*	*Olympia.*
Type of pump.....	Crank shaft.	Direct driven.	3 throw crank-shaft.
Space occupied by pump, cubic feet.....	299.	166.5	311.
Floor space, square feet.....	15.3	16.25	19.25
Space occupied per cubic foot displacement of pump, cubic feet..	35.8	22.9	28.5
Floor space per cubic foot displacement of pump, square feet.....	1.8	2.2	1.7
Space occupied per I.H.P. main engines, cubic inches.....	156.	114.	62.
Floor space per I.H.P. main engines, square inches.....	.7	.91	.32

In a yacht engine that I designed last year to make 300 revolutions per minute, and of 200 horse power, I placed the air pump in its usual place in the main engine structure, and it has given no trouble, the engineer being fireman also, has little time and less skill to bestow upon an air pump that does not attend promptly to its duty.

I am thankful to Mr. Bailey for the way in which he has presented this subject, and trust that he will be able to present its economic side in another paper.

[The discussion of this paper will be continued in the November number of the JOURNAL.—*Editor.*]

ELECTRIC TRANSMISSION OF POWER AND ELECTRIC LIGHTING, FROM THE POINT OF VIEW OF THE MECHANICAL ENGINEER.

By Professor Alexander B. W. Kennedy, LL.D., F.R.S.

[Address delivered before the British Institution of Mechanical Engineers.]

It is a well-known difference between engineering practice in this country and in most others, that the conditions of our work seldom allow us to carry specialization to any such great extent as is common, for instance, all over the Continent. With us the belief that a man is "a thorough practical engineer"—whatever that may mean—is very generally taken as evidence that it is safe to ask him to do anything whatever which may be included within the vague general term "engineering." Of the extent to which this feeling is carried, as well as—I must say—of the essential truth which lies at the bottom of it, no better illustration could possibly be found than the position now held, with so much advantage to the public service, by my distinguished predecessor in this chair, Dr. Anderson. Some of us have occasion to examine continually the epitomized professional histories of engineers who wish to belong to, or to be promoted in, our Institutions. It is quite noteworthy in how many cases, especially in our own Institution, a man of thirty-five or forty is found to be working with great success in a line totally different from that in which he gained his first experience.

It happens that, having served my time with marine engineers, I have had in the last twenty-five years to busy myself with half a dozen varieties of mechanical and structural work (including what I may call the academic variety during many happy years at University College), and of late years also with the newest

variety of all—electrical engineering. I trust that my electrical friends—those of them, that is, who are twenty years younger than I am—will forgive me for calling electrical engineering a variety of mechanical work. I mean nothing disrespectful, of course; but even to those who are much more thoroughly electrical engineers than I can ever hope to be, but who have had the chronological misfortune, like myself, to have become engineers before technical electricity existed (if I may be allowed so to speak), I think it must always be so. To such of us, electrical work has been grafted on to mechanical; and any success we have had in the one, we trace back to its foundation in our knowledge of the other. And further—although I know that in this I lay myself easily open to an accusation of mere conservatism—I believe that even now the only really safe road to success in electrical work lies through the old routine of mechanical training, modified, of course, by the excellent means for studying the scientific side of the work which now exist in so many directions.

Practical electrical problems divide themselves probably into three main sections, in which electrical energy is used respectively for:

1. Lighting.
2. Power.
3. Physico-chemical processes.

The third section—where energy is used for the deposition of metals, for the reduction of chemical compounds, &c.—falls outside of my own knowledge. From what I know of the success with which it is being worked, I cannot doubt that it has a great future. But its problems only touch engineering very indirectly as yet, and, therefore, hardly come within the subject which I have prescribed to myself. In this section may be included, at least for present purposes, the application of electricity to heating. Not only the difficulty of producing the heat, but also that of producing it in a convenient fashion, have been overcome to a very large extent. The remaining difficulty is the purely commercial one of producing the heat sufficiently cheaply to allow of its general use. This, I am afraid, has not yet been overcome;

and until it has been, electrical heating unfortunately hardly comes within our ken, except for very special purposes.

Remembering that something like 95 per cent. of all the energy that goes to incandescent lamps appears only as heat, and not as light, there appears to be an ample opening here for another "thermal storage" process, at least until the present lamps are superseded by the less wasteful type which the future will doubtless reveal to us in good time.

The use of electrical energy for power, *i. e.* for transformation into mechanical energy, is a matter which lies obviously in the closest possible communication with mechanical engineering. It divides itself broadly into three sections, viz: (1) Transmission from a distance for whatever purpose, (2) Transmission to a number of isolated points comparatively near together, as the tools in a factory, and (3) Transmission for the purposes of traction on railways or tramways.

The *transmission* comes in in every case because we have as yet no electric prime mover analogous to a steam engine. The electric motor presupposes the existence of a dynamo, which must itself be driven by some prime mover. From the dynamo to the motor, whether near or far apart, there must always be transmission of electrical energy. In a recent extraordinary aberration it seems to have been discovered that there was some positive advantage in using steam engine, dynamo and motor all in one place to do work hitherto done by the steam engine alone. This wonderful locomotive, however, forms, and no doubt will continue to form, a class by itself!

The question of driving the tools in a factory by electric motors instead of through counter shafting is one which has recently come to the front. The carrying out of such work is obviously purely a matter of mechanical engineering. Its advisability in any particular case is partly a question of economy and partly one of convenience, and is undoubtedly dependent on the conditions of each particular case. I have found from information very kindly given me by several large engineering firms that the actual cost to them of power, including coal, stores, wages and

depreciation, generally lies between 2 and 5 per cent. of their total cost.

In any given factory running on the ordinary system, there is a large continuous waste of power due to the running of the whole shafting, no matter how many or how few machines are at work. Under such conditions the waste work in shafts and belts may well be even 25 per cent. of the average useful work, and the distribution of total work may be approximately:

	Horse power.
Average useful work,	100
Wasted in belts and shafting,	25
Wasted in engine friction, the engine being supposed large enough to give 150 horse power at tools as a maximum (at about 10 per cent. of maximum horse power),	20
	145

Now, if all the machines in such a case were driven by separate motors, each having an electrical efficiency of 88 per cent., and these motors worked from a dynamo having an efficiency of 92 per cent. (both of which are high figures for ordinary work at two-thirds output), the figures would stand as follows:

	Horse power.
Average useful work,	100
Wasted in motors and dynamo,	24
Wasted in leads (say 2 per cent.),	2
Wasted in engine friction (as above),	20
	146

It will be seen that the two sets of figures are practically the same as to amount of power required. As the electrical efficiencies which I have assumed are not likely to be exceeded, I think it may be said that there is no saving to be obtained in horse power, and none, therefore, in any of the items of costs directly dependent on horse power, in cases where the power wasted in shafting and belts does not exceed 25 per cent. of the whole

average useful power. This is a ratio which any engineer can most readily obtain for himself. I have no doubt cases can be found where the waste in large factories with very widely distributed power is very much greater than the proportion assumed, and in such cases no doubt a saving in power costs would occur.

But at the root of the matter there probably lies one ultimate determining point, far too often neglected in such calculations as I have just been making. This point is, of course, the real absolute saving to be made as distinguished from the proportionate saving. It is of little use to show an engineer that he can make even 20 per cent. saving in some one item of expenditure if that whole item only represents 3 or 4 per cent. of his costs, and if at the same time he has to expend a considerable amount of capital in making the change. In reference to our present point, it will prove sufficient to prevent the adoption of isolated motors in engineers' works in all cases except where the waste in belt and shaft transmission is exceptionally great, or where, for other reasons, electrical driving *per se* is desirable. For the whole matter is certainly not concluded either one way or the other by this single consideration.

There can be no doubt, on the one hand, of the practical convenience of getting rid of the huge mass of shafting, gearing and belting, which fills up the upper half of many engineers' shops, although it is difficult to fix any money value for this advantage. There is no doubt also that a properly arranged motor may give a much larger range of speed to each tool than can be readily obtained in the ordinary way, and will thus enable more and better work to be got out of it with skillful management. Were it not that such magnificent work has often been turned out in this country from such dirty and apparently disorderly shops, I should also say that it was self-evident that every change which improved the general orderliness and tidiness of the shop must—reacting upon the men employed—improve the output both in quantity and quality.

On the other hand, it has to be pointed out that the cost of dynamo, leads and motors is very greatly indeed in excess of

the cost of shafting in almost every case. It is also hardly certain as yet how the costs of attendance, lubrication, renewals and repairs to the electrical plant, compare with the similar costs in the case of shafts and belting. Probably on the whole they would be less, but no satisfactory data on this point exist.

There arise two special cases which should be mentioned. In cases where electrical energy can be obtained from public mains the comparison is quite different. The working of the printing machines, for instance, in the office of the "Pall Mall Gazette" is carried out electrically, the current being derived from the mains of the Charing Cross and Strand Electric Supply Company. The works in this case get rid of all costs of engines and boilers, with all their many inconveniences, and also gain in space most of the room occupied by their old plant. Moreover, they have a better security against breakdown than they could otherwise very well afford to have without most costly duplication of plant. The matter then becomes simply one in which increased space, convenience and security are to be balanced against increased cost, and each case has to be considered on its own merits.

At present it is not very certain how the engineer supplying the electric energy ought to look at such a case. His chief business is, and will be, to supply energy for lighting purposes. As is sufficiently well known, the demand for lighting energy varies enormously throughout the twenty-four hours, so that, in fact, a plant which is giving 2,500 horse power for a couple of hours every day, and which, of course, must be large enough for this purpose, leaving still machines in reserve, will only be giving an average of 350 horse power taken over the whole week, and not even half this for many hours every day and night. It is, of course, very desirable to increase the load during the hours of light load, and if this could only be done without also increasing its present maximum, that is, without necessitating increase of plant, engineers could afford to sell the additional energy at a very low rate indeed. Unfortunately, the condition I have mentioned is impossible. There will always come hours when, through fog or other special causes, the power load will coincide in time with the

lighting load. And even if this were to happen only once or twice a year, the engineer must none the less provide plant for it and keep the plant ready. The case is not met by differential charges at different hours. The plant must be provided all the same, and the additional outlay will not be met by the extra payment for a few hours' work per annum. It may, however, very well be met by the value of the additional work done on 360 days out of 365 at times when otherwise nothing would be done. This point I shall deal with later on.

A second special case occurs when the cost of power forms a very large instead of a very small proportion of the whole running costs of the works, and no doubt in several industries this is the case to a very much greater extent than in engineering works.

I will ask you particularly to notice that in the case where electrical driving supersedes belts, the work is not done any less directly than before. Between the engine and the tool there stand, in the one case, belts and pulleys and shafts twice or thrice repeated, and in the other case dynamo and leads and motor, with or without a small belt from motor or machine. It is quite possible that the actual mechanical efficiency of the belts is greater than that of the two electrical machines and the leads between them. But in the case of the shafts and belts the losses are continually going on whether or not useful work is being done, whereas there are no losses in the motors unless they are actually running and driving tools. This brings the total losses equal in the case I have supposed, and in other cases may make the electrical method more economical. But the change is merely a substitution of one indirect method for another equally indirect method.

In the third great case of electric transmission of power, for the purpose, namely, of driving trains or tram cars, we have a state of affairs which is entirely different, and which, I am afraid, is by no means so favorable to the use of electricity as is sometimes supposed. In the case of a train drawn by an engine in the usual way, we have the whole of the net power of the engine, exclusive of that spent in internal friction and in actually haul-

ing itself along—say 80 per cent. of the gross indicated horse power—available for pulling its load along the line. In the case of an electric railway, we have truly a somewhat lighter locomotive to be moved, but against this advantage and the fact that a stationary engine can have a greater economy than a locomotive, we have the uncomfortable state of affairs that only some 35 per cent. of the whole indicated horse power, as against 80 per cent., is available for the useful work of pulling the train. It seems, moreover, hardly likely that this low percentage will be much increased. The cause of the loss is, of course, the extreme indirectness of the process, that is, the very great number of transformations through which the energy has to pass, in each of which there is always a loss. We may take the figures approximately somewhat thus, always remembering that the average power would be less, probably very much less, than three-quarters of the full power of the plant at work :

	Per cent.
Mechanical efficiency of engine, ...	85
Efficiency of belt-driving (if employed), . .	94
Efficiency of dynamo, ...	90
Efficiency of line, ...	85
Efficiency of motors, ...	85
Efficiency of gearing on locomotives, . . .	75
Total efficiency, ...	39

From this is to be deducted, say, 10 per cent. for driving the locomotive, leaving 35 per cent. for pulling the train.

I think it may fairly be said that, to a mechanical engineer, the greatest enemy to all economy is frequency of transformation, and that it is, or ought to be, a fundamental rule in all engineering work that every unnecessary transformation, whether mechanical or electrical, should be avoided.

Notwithstanding this terrible drawback, the absence of foul gases is of such overwhelmingly great importance in some cases, that the working of the City and South London Line has only been possible at all because electricity has been employed instead of steam. This, of course, is a case where mere economy

of power was not the only or even the most essential thing sought for. In the Liverpool Overhead Railway, also, the conditions of the case rendered the use of steam locomotives impossible, as the line runs along a route on which they are prohibited.

The case of tram lines along streets differs altogether from that of an underground or indeed any other railway. There is probably no form of steam car which would be generally endured by the public. Here again, therefore, electricity would appear to come in regardless of cost, in the same way as with the underground railway, although for another reason. But here we are met at once with the mechanical difficulty (not existing, of course, in the tunnel or on the overhead railway) of getting the current to the motors on the car. In America the knot is cut, rather than untied, by the use of overhead wires, and in country places it is possible that this may be here the best solution of the problem. But in our cities, where the bulk of the tramway traffic of the country is really carried on, I am still conservative enough to think, and also to hope, that this is impossible. Without forgetting the much-quoted case of Buda-Pest, therefore, I fear that the introduction of electricity for car driving in this country will still wait until a practical underground system has been devised. Meanwhile it is being hard pressed by its rivals—cable and compressed gas. Of the two I think the latter, although the younger, is far the more formidable. It has the advantage of being even more direct than a steam engine (the boiler being absent); it can be applied to each individual car even more easily than an electrical motor, and it enables the car to run freely on ordinary lines without their reconstruction, and without any mains either above or below ground. It has as yet had but short trial, but what I have seen of it makes me sanguine as to its ultimate possibilities.

Before speaking more particularly about the mechanical aspect of electric lighting problems, I feel bound to touch upon a matter which arises directly out of the question of supplying electric energy for power and for light concurrently, to which I have just drawn attention. It is a favorite bit of cheap criticism indulged in by those who have not to supply energy on a large

scale at the expense of those who have, that the former class of folk show their ignorance of their own work and interests by neglecting to obtain that enormous demand for motive power which would come on them if only they reduced their prices low enough. The criticism is as really absurd as that other which censures the gas companies because they do not supply heating gas at a few pence per thousand feet, which apparently they would be perfectly able to do if only they knew their own business half as well as their critics do! In both cases the criticism is mere words, "signifying nothing." We have simply to remember the fact that the mere possibility of driving something by electricity does not of itself create the things that want to be driven. There are isolated districts in London in which there is a considerable amount of power of a kind which might be furnished electrically if the cost were made sufficiently low. In most parts, however, the whole energy expended in doing mechanical work is but a very small percentage indeed of the energy required for light, and it would not be increased in the least even if it were supplied for nothing.

Whether in a purely manufacturing district it may ever be found to pay to put down a central power station and transmit power to factories, each of which would replace its steam engine and boiler by an electric motor, is a question about which there is a good deal to be said on both sides. I have already indicated the lines along which I believe this matter will settle itself.

I pass now to the last and largest section of my subject, the question of public electric lighting from the point of view of the mechanical engineer. From the side of the public, that is, of the users of electric light, the essential points are simple, viz., that the light should not visibly flicker, that it should not be below the declared pressure nor very much above it, that it should be available at all hours, and of course most of all that it should never go out! But any notions one might have that in consequence of the simplicity of this final result the means of attaining it would also be simple, are very speedily dispelled by a consideration of the facts. It is, alas! not sufficient to provide a shed, a boiler, a decently governed engine, and a dynamo or

alternator, and many and sad have been the disappointments in consequence. The fact is, that although the final result to be attained is always the same, and is always extremely simple and straightforward, the conditions under which that result has to be obtained are always varying, and are not only extremely complex in any one place, but are also different altogether in each different place.

I will consider first those variations which are entirely beyond the control of the engineer, *i. e.*, the variations of load, and afterwards those which are within his power, *i. e.*, variations in the working of the station. To those of you who have practically to do with these matters, the variations I have to speak of are absurdly familiar, but to others—probably the majority—they are still sufficiently unfamiliar to be of some little interest. I have, therefore, prepared diagrams which may illustrate them.

Diagram 1* shows the variations of output of a lighting station during a whole year. Each vertical division stands for one week. Irregularities are due to particularly foggy weeks, or to weeks in which public holidays occur, such as Easter and Whitsuntide. For the rest the curve follows pretty obviously the changes in daylight, except that the minimum occurs not in the very middle of the Summer, but during August, in consequence of the large number of private houses which are shut up at that time, while the days are still too long to require the use of any artificial light in shops which close at seven o'clock.

Diagram 2 shows the variations within a week, each vertical division standing for one day. Here, of course, the chief variation is that which occurs on Saturday and Sunday.

Diagram 3 shows the variations occurring within one day, the vertical divisions each standing for one hour. In this diagram three days are actually shown, one in December, one in August, and one in June, and it will be noticed that the three are very

* This diagram corresponds to the output at the Davies-street station of the Westminster Electric Supply Corporation, during 1893. The great excess in the last quarter over the first quarter is to a considerable extent due to the larger number of lamps which were on circuit towards the end than at the beginning of the year. The great drop in the last week of all is due to the fact that that week included Christmas day and Boxing day as well as a Saturday and Sunday.

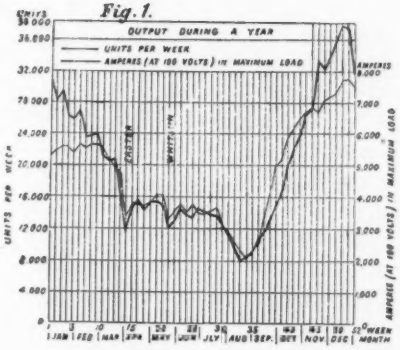


Fig. 1.
OUTPUT DURING A YEAR
UNITS PER WEEK
AMPERES (AT 100 VOLTS) IN MAXIMUM LOAD
UNITS PER WEEK
AMPERES (AT 100 VOLTS) IN MAXIMUM LOAD
EASTER
WHITSUN
1 JAN
8 FEB
10 MAR
12 APR
20 MAY
22 JUN
30 JULY
35 AUG
35 SEP.
42 OCT
43 NOV
50 DEC
52 WEEK
MONTH

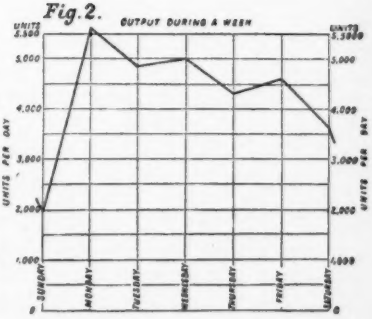


Fig. 2.
OUTPUT DURING A WEEK
UNITS
UNITS PER DAY
SUNDAY
MONDAY
TUESDAY
WEDNESDAY
THURSDAY
FRIDAY
SATURDAY

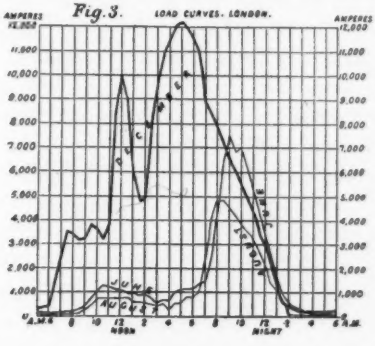


Fig. 3.
LOAD CURVES. LONDON.
AMPERES
DECEMBER
MARCH
JUNE
AUGUST
NOON
MIDNT

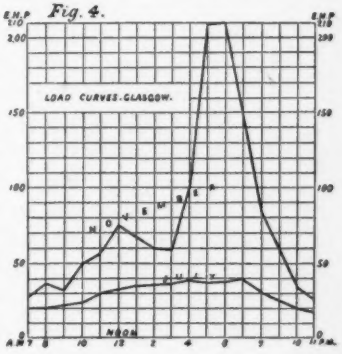


Fig. 4.
E.H.P
LOAD CURVES. GLASGOW.
NOVEMBER
JUNE
NOON

different. In winter there is practically always a double maximum, one occurring about breakfast time or in the forenoon, after which the load falls off in the middle of the day and reaches its full maximum towards evening. In the summer time the morning load disappears altogether, and there is only one maximum, which occurs much later than it does in winter. In fact, I have noticed that in London at any rate, the average time up to which people use light is fully an hour later in summer than it is in the winter, a somewhat interesting fact in sociology. In August the "peak" of the diagram becomes a minimum, the private houses using no light because they are closed, and shops because the days are still too long.

The thin line on Diagram 1 shows in rather an interesting fashion that the maximum load and the maximum output by no means vary together. The maximum load is much more constant than the maximum output, and does not vary nearly so fast or so much. The quantity which chiefly varies is the time during which that load lasts, and this time is, of course, dependent on the hours of sunlight, or more particularly on the interval between sunset and closing, or bedtime.

Besides the variations shown in these diagrams, it is right that I should mention one or two others. A very important class of variations are those which may be said to be due to latitude. In a place as far north as Aberdeen, for instance, there is practically no night for two months in the year, and for a much longer time than this no lighting whatever is used in shops, and hardly any in private houses. On the other hand the hours of lighting in the winter are proportionately longer than in the south. The effect upon the annual output diagram would be not to alter its area considerably, but to thicken it up in the winter and reduce it very much in the summer. As an illustration of northern loads I have shown in Diagram 4 load curves for a winter and a summer day, taken at a little station which was running in Glasgow supplying offices some two years ago.

One set of variations which puzzles and troubles the engineer very much is the varying amount of light thought necessary by people in different parts of the country. No doubt this depends

to a considerable extent upon the nature of the premises in which the light is used, but even this is hardly sufficient to explain the whole of it. In a purely residential district in London, for instance, the average energy used per annum is only about 12 units per lamp wired. In a mixed district like that of the Westminster Company it is about 16 units per lamp wired. In a very rich district containing many clubs, like that of the St. James' Company, one is not surprised to find that it amounts to 22 units per lamp wired. But one is on the whole surprised to find that in Newcastle it should be as much as 25 units, and in Leeds 27. It is no doubt with less surprise that one notices that the consumption of 38 units per lamp which existed in Exeter when current was paid for by contract and not by meter, was exactly halved as soon as consumers were made to pay for the amount which they really used.

There is one set of variations which, unlike all those which have been mentioned above, is to a certain extent under the control of the engineer. I mean the variations on the two sides of a circuit which is arranged upon a three-wire system. Here, of course, the engineer in the first instance endeavors to connect about the same number of lamps wired on each side of his system, and to see that the balance is approximately right, not only over the whole system, but over each small section of it. In spite of all the care that may be used in this matter, however, I have found that a three-wire system may be very often greatly out of balance for a few minutes at a time, owing to causes entirely beyond the engineer's control. Rapid changes in this respect occur chiefly within the fifteen or twenty minutes during which load is coming on most rapidly in the afternoon. In a station when the load was increasing from 2,000 up to 8,000 ampères, I have frequently seen that in the course of ten minutes first the positive and then the negative side would have 400 or 500 ampères more than the other. By the use of a balancing machine which can be transferred by a single switch from one side to the other, the case of course can be met as it arises, but it is sometimes a little surprising that the changes should be so large and so rapid.

I have endeavored to indicate to you some of the great number of variations in conditions, in spite of which the engineer has to obtain a uniform result. All these variations—even the last—are practically altogether beyond his control. Some of them occur with considerable regularity, and can be foreseen; others are most irregular, and come without any warning whatever. But all of them affect very greatly both the method and the cost of production. The method first, because the efficiency and regularity of supply must be considered before the matter of economy. Any method of supply which was inevitably liable to breakdowns and interruptions would be wholly condemned on that ground alone, even if it happened to be of the very highest economy. But given any one of the many methods of supply which are equally good as to result to the consumer, then that method must mainly be judged by its economy under the particular conditions in which it has to work. It will, of course, be borne in mind that the cost of power production in this case is not a mere 3 or 4 per cent., as in the case of a factory. It is 50 or 60 or 70 per cent. of the whole expenses, the balance being expenses of management and distribution, along with interest, depreciation, and other charges connected with capital.

As to the effect, then, of the uncontrollable variations on system or economy, the first thing which I have to point out is that their madness is not entirely without method. There are two points of special importance. First of all, there will never be more than certain proportions of all the lamps which are wired alight at once, either at times of maximum or of minimum load. The maximum maximum on the circuit of the Westminster Company is about 35 per cent.; on that of the St. James' Company, I believe, over 60 per cent.; of the House to House Company, about 35 per cent. In Glasgow, where a present the light is solely in the region of offices and shops, the proportion is as high as 56 per cent. You will see that these proportions depend largely upon the nature of the places where the light is used. This proportion, which, of course, can be only an estimate in any particular place until the experiment is

made, fixes for the engineer the maximum number of lamps wired for which a given horse power will supply current.

The maximum minimum is an almost equally important quantity, and in most places the engineer has to do with two such maxima, one for the day hours and another for the night. In a purely residential district the night load will always be very small, and the day load will be practically *nil*. In the City of London the day load, in consequence of the number of dark places to be lit, is considerable, the night load extremely small. In a mixed district like that of the Westminster Company, the day load in the middle of summer is not greatly more than the night load.

It is to be noted, however, that the night load is practically the same all the year round, while the day load, even apart from fogs or exceptional weather, necessarily varies very much. The maximum maximum also, as we have seen, varies very much at different times during the year, although it never exceeds a certain amount. It is also worth noticing that in general the night load on a central station will increase but slowly with the growth of the station. The first consumers who take the light are most often those who are most in need of it, who require it for dark corridors and long hours. Later on thousands of lamps may come on to a circuit in shops, offices and in ordinary private houses without a single one of them being used after midnight. In an average district the day load increases faster than the night load, because out of any large number of shops and offices some are sure to have dark places requiring current in the daytime. But neither night nor day loads increase nearly as fast as the maximum evening load, which, a certain normal condition having once been reached, will increase exactly in proportion to the lamps wired. In an alternating current station where very numerous isolated transformers are used, the night load increases nearly proportionately to the day load, because of the increase of magnetizing current.

The determination of the sizes of units to be used in any station depends essentially on the probable values, absolute and proportionate, of the loads I have mentioned. It is now a commonplace, although a few years since no one seems to have

thought of it, that the machinery in a station should be so subdivided that under all sets of conditions which remain steady for a few hours at a time it should be possible to have such machinery as is running, running more or less nearly fully loaded; only in this way can much economy be hoped for. It must not be taken, however, that this is a mere question of arithmetic, which can readily be settled off-hand. In every case—it is to be hoped—the output of a station will be much greater ten years hence than it is to-day. The engineer has not uncommonly to consider somewhat difficult *pros* and *cons* when he is called upon to provide machinery which shall be of such a type that it can allow dividends to be earned during the years of growth and yet remain still suitable in the time of maturity.

It is further to be remembered that this question of the choice of size of unit is also quite modified in the cases where condensing plant is used instead of non-condensing. A good condensing engine uses nearly the same steam per indicated horse power hour at any load above half output. The increased consumption per electrical horse power hour is therefore proportionately much less than in the case of non-condensing engines, and the economical importance of fully loading the machines is correspondingly minimized in stations where condensing plant is employed.

The question of size of unit leads naturally to that of the necessary magnitude of reserve in any case. I think this is a simple matter, but no doubt it is sometimes wrongly treated. To make the reserve a fixed proportion to the total plant, for instance, is a method which can only possibly come right by a fluke. I will not argue for the necessity of a reserve—that appears to me to be beyond any argument. The reserve is required for two purposes—to provide against breakdown of plant, and to provide against accidental increase of load on special occasions. It is in the primary conditions of the case that the light absolutely must never go out, even if a machine wholly breaks down at the time of full load. From this it follows as a mere matter of arithmetic that the minimum of reserve must be a unit equal in size to the largest unit in the station, so as to be capable of taking its place if it breaks down.

Those questions about the running of a lighting station which are essentially matters of mechanical engineering, group themselves mainly in three sections, according as they relate to (1) security, (2) efficiency of regulation, and (3) economy of working. It is obvious that the fundamental question of the choice of an electrical system in any particular case connects itself with all three.

Of the three points I have put security first, because no doubt the absolute certainty that nothing short of a boiler explosion or of the burning down of a station shall stop the supply of current is really the very first condition of supply. Clearly with any machinery the burning of an armature, the seizing of a bearing, or the breaking of a shaft, are possibilities which may have to be faced at any moment. They seldom occur—very seldom, happily—but sooner or later one or other will come about, and the engineer in charge has to be provided with means for preventing any such accident having more than the most transient effect on his light. Of course, the existence of proper reserve plant, such as I have already mentioned, is the first necessity of the case. Scarcely less important, however, is the necessity that such reserve should be almost instantly available. Where all the plant is running in parallel, *i. e.*, where all the dynamos or alternators are working straight through the same pair of omnibus bars on the switchboard, it will very often happen that the other machines which are at work may of themselves be able, at least for a few minutes, to take up the load of the one which has broken down, and so to allow time for another machine to get started. It may quite well happen, however, that these machines may be themselves running very nearly at full load, and that the broken down machine may have had so large a load upon it that this is impossible. In any case a large machine, say 400 horse power or more, cannot be started and got on circuit in a few seconds, and seconds are of importance in these cases. For this reason, in stations which use alternating currents it is usual, and probably necessary, to have one reserve machine always warmed up and running round at half speed, so that it is merely a matter of opening a stop valve to let it away in a condition ready for put-

ting on circuit, although the synchronizing in is unfortunately always a matter which may cause some minutes' delay. In continuous-current stations it frequently happens that a battery is used which can give so high a discharge for four or five minutes as to take the place—along with the machines remaining on circuit—of the machine which has given way, until another machine has been started and put on. So far as it goes this is no doubt a great advantage, but it must be remembered that the mere existence of a battery, and even of a battery quite sufficient for helping in day and night light loads, is not sufficient to insure that the battery shall be available for help in this fashion. Two details are easily seen to be essential when it is remembered that the failure may occur during the hours of full load. As the volts are highest at such a time the battery must not only have sufficient capacity, but must also have a sufficient number of cells to give in discharge the full electromotive force of the station, and this will not be the case unless the engineer has specially provided for it by an unusually ample number of cells. Secondly, the battery must have been fully charged beforehand, so as to have the maximum electromotive force per cell available. This involves that the battery charge should have been completed during the day, before the hour at which full volts could possibly be required from it in case of any accident. A battery which could only discharge heavily at 215 volts would be practically no use whatever as security against a drop of pressure after an accident which might possibly occur at 230 volts.

In an alternating-current station where the machines are not run in parallel, but where each alternator supplies current to its own separate circuit or bunch of circuits, the sudden failure of a machine must instantly cause extinction of light in the circuit affected for the fraction of a minute which must elapse before the circuit can be thrown on to other machines, or the few minutes which it must take to get another engine started and plugged on. In a station worked on this plan, however, the machine which is turning round as stand-by can be kept excited and giving the volts corresponding to its speed, and can be plugged on in readiness, so that the circuit from any machine which has

broken down can be instantaneously switched on to it and it can be run up at once to full electromotive force without, of course, any of the delay which would be unavoidable if it had to be synchronized and put in parallel. It has also to be borne in mind that in cases where a number of alternators are running in parallel, the breakdown of one may tend directly, in certain cases, to endanger all the others—a trouble which does not exist if circuiting be used. On the other hand, the circuiting system causes inevitable, if slight, flickers in the light, which are quite avoided by running in parallel. It also necessitates the use of comparatively numerous circuits, as no one can be allowed to carry more than from 30 to 50 kilowatts at a time when it is switched on or off.

It must be recognized that in one matter the requirements of security and of economy are more or less opposed. If an engineer is quite certain that whether by the help of batteries or otherwise he can bring his reserve into action so promptly that the worst breakdown will not spoil his lighting, he will no doubt always run each unit of his plant as nearly at its full capacity as possible. But if for any reason he cannot be so sure of his reserve, he will gain greatly in security by running his plant always so far under its maximum that he can distribute the load of any one unit for a few minutes over the others without endangering them. But of course this is practically equivalent to underrating the whole capacity of the plant by a considerable percentage, probably 20 per cent., and consequently of increasing the capital cost of the plant per unit of normal maximum output by just the same percentage—a result which one does not wish to face.

I have spoken of possible breakdown of engines or dynamos. There are, of course, other accidents to be provided against. Short of a boiler explosion, the most serious of such accidents would doubtless be one occurring to a steam pipe. The physical consequences of the actual bursting of a steam pipe are not, of course, the matters at present under consideration. In all probability such an accident would disorganize any station, no matter how well arranged. But short of the actual bursting of a pipe, many things may happen which may render it necessary

to shut down a certain portion of piping. It is generally necessary, therefore, to provide any lighting station with something in the nature of a ring-main system, or its equivalent, by which any engine in the station can be supplied with steam from two directions. In a properly arranged ring-main system it should be possible to disconnect or cut out of use any section of steam pipe which may be giving trouble without stopping a single engine. Of course it is not necessary to provide pipes so large that under these conditions there should only be the normal drop of pressure between boilers and engines. One can put up with an extra drop without grumbling if it only occurs under these circumstances. It is inevitable, to reach the required degree of security in this matter, that the steam pipe should be fitted with a very large number of valves, and these are undoubtedly objectionable, if only on account of the very considerable loss by radiation, which is almost inevitable from their covers and flanges. But, after all, the making of good steam joints in large pipes at 150 pounds pressure is nowadays a very simple matter, and the unavoidable complexity of the arrangement is more than counterbalanced by its usefulness.

The duplication of the steam pipes must obviously extend to the boilers as well as to the engines. The hanging up of a safety valve, the blowing of a mudhole joint, the bursting of a tube, or any one of half a dozen other such accidents, may put a boiler suddenly *hors de combat*. It must be possible, therefore, at the shortest notice, to cut out any boiler without interfering with the working of the rest. Fortunately, boilers can be forced more readily than engines or dynamos, and such an accident could almost always be met by forcing the remaining boilers somewhat.

Reserve in boilers stands thus in the same position as reserve in machines, except in so far that the boilers can be more easily forced, as I have just pointed out. With them, however, as regards security of working, there comes in an even more difficult question than that of reserve, namely, that of readiness for emergencies other than breakdowns. Even in London, where a fog may cause the load on a station to increase 10 times in as

many minutes, there is practically no difficulty in getting the engines started quickly enough. It is otherwise with the boilers, which, although so easily forced when actually in work, are terribly sluggish in getting up steam, as compared with the rapidity with which unexpected darkness comes over London. At the most inconvenient time for a breakdown, namely, the time of full load, the engineer knows at least that an engine or dynamo accident can hardly affect him to more than 15 or 20 per cent. of his whole work, of which, at the worst, a considerable portion can in most cases be taken up by the remaining plant. But the most inconvenient time for a fog to come on is at light load, and then the increase coming on in a few minutes may be anything from 500 to 1,000 per cent. And the engineer must be able to meet this in June as much as in December, and with even less notice. Practically this means that, even in summer, enough boilers must be kept under steam—if not at full pressure, at something near it—to be able to take up this enormous increase of load in 10 minutes, as soon as ever their fires are brought forward and their dampers opened. I need hardly point out to you, after going over all these points, that it is improbable that the fuel consumption per indicated horse power hour in electric light stations will ever reach, or can ever reach, anything like so low a figure as has often been reached in first class steamers, where the engines and boilers work on continuously day and night at pretty much their most economical load.

I will mention just one more point as to security of working in a station—the feed arrangements. A duplication of feed pumps and pipes is an obvious precaution, and one that probably every one adopts. But few of us can keep water stored in sufficient quantity to last for more than three or four hours at full load, so that without special arrangements we are liable to be extremely inconvenienced by any accident happening to the mains of the water companies outside, which may compel them, on short notice or even no notice at all, to cut off the supply for a few hours. The best arrangements to meet this contingency vary in different places, but the engineer must not lose sight of

the fact that the difficulty is one which may occur at any time, and from causes altogether beyond his control and even knowledge.

Outside the station the question of security comes in in connection with the mains. To a certain extent every station which uses either high or low-tension feeders has some reserve in their very number. When a company has more than one station, trunk mains led from one to the other are also a considerable security, as well as a great help to economy. Mr. Frank Bailey, of the Metropolitan Electric Light Company, has carried out a most excellent ring or looped main system on his high-tension distributors, by which he can make sure that any consumers can be supplied two different ways round the circuit, so that any accident to the mains may cause a minimum of inconvenience to a minimum number of consumers. In ordinary low-tension distributing networks, the same result should be aimed at, although it cannot generally be obtained in what the mathematicians call so "elegant" a manner.

I have next to speak of the *regulation* of an electric lighting station, as forming, along with its *security*, the two matters which chiefly affect the consumer of electrical energy. All that the consumer cares about is that the pressure in his house should be steady to the eye, and should be not less than its nominal value, and not very greatly above it. In all stations working on a large scale this constancy of pressure in the consumers' houses necessitates a constantly varying pressure at the station, in order to make up for the loss of pressure in the mains, which is itself —wholly or in part, according to the system of supply—dependent on the amount of current passing through them. In an ordinary alternating-current station the pressure at times of maximum load exceeds that at times of minimum load by about 5 or 6 per cent. In an alternating-current station, with low-tension distributors, the excess may be about 12 or 15 per cent. In an ordinary low-tension station, with three-wire system, it may often be 20 per cent. The actual figure, of course, depends on the current density permitted in the copper; but the figures I

have given illustrate the sort of percentage which different engineers have found it advantageous to adopt on a large scale.

The primary requisite for proper regulation is obviously enough that the engineer shall know the actual value at each instant of the pressure which he desires to keep steady. In every case where low-tension distributing mains are used, this pressure is the pressure at certain feeding points or at certain sub-stations, and it is best known at the central station by direct indication—bringing back pilot wires or potential leads from such points direct to the station and connecting them to suitable voltmeters there. In these cases the regulation consists in so running the machines that the pressure indicated by these voltmeters is always kept at the desired amount. What appears to me a less satisfactory method is sometimes adopted, namely, the determination by calculation, or by experiment, of the loss of pressure corresponding to each output, adding this loss to the normal pressure, and regulating the dynamos to the corresponding sum, transformed if necessary. Apart from the fact that in this case the pressure actually received by the consumer is only estimated and not really measured, so that the thing measured is not the really important quantity, there is the additional drawback that the regulation is to a varying quantity instead of to a constant one, and is, therefore, less likely to be thoroughly satisfactory.

In cases where distribution is by high tension, with isolated transformers, there is, of course, no direct means for keeping the pressure constant in the consumers' premises. If the distance between the nearest and the furthest consumer on any circuit is much less than the distance from the station to the nearest consumer, there may be a reasonably good regulation by the indirect means I have just described. In the case where the distance from the station to the nearest consumer is small in proportion to the distance from the nearest to the furthest, no such regulation is practically possible; and, in fact, in these cases, the section of the copper must be made so large as to reduce the drop of pressure within negligible limits.

By one or other of the methods which I have mentioned, and

preferably, no doubt, by the use of pilot lines, the engineer in the station knows to what pressure he is to work. In every case the station must work at a varying pressure, although in some instances the variation may not be great. The highest pressure must occur always at the times of greatest load. Several means are at the disposal of the engineer for varying this pressure. Best of all, I think, as being most convenient and most easily controlled, is the provision of sufficient additional resistance in the fields of the dynamos or exciters to enable the magnetic field to be altered by the mere moving of a switch having many contacts. Of course, to a certain extent this method wastes energy, but the quantity so wasted is so very small in proportion to the whole, that it is not too high a price to pay. The switches can be placed upon the switch gallery so that the whole of the regulation may be done by the man in charge there, and be quite removed from the actual control of the engine-driver. In order that this method may be satisfactorily carried out, the engines must be so arranged that they can be run entirely upon their governors, the stop-valves being opened fully as soon as the machines have taken up any load. In this connection it is worth while noticing that for continuous-current machines constancy of speed—which is, of course, to be distinguished from steadiness of speed, quite a different matter—is not in itself in the least a thing to be insisted on. The governor must be so set as entirely to prevent the engine racing in the case of a sudden short circuit or other breakdown. But apart from this, an alteration of 5 per cent. or more is not a matter of the least moment. In fact, for practical purposes, it is generally better that the governor should not be too finely set. Of course it may quite well happen that in the case of any one engine it may have to run throughout nearly its whole range of power at the same electrical pressure. If, therefore, the governor is so arranged as to allow of a drop in speed at full load, the field resistance must be so proportioned that notwithstanding this drop it can either keep the pressure constant or allow it to be increased within the desired limits.

A much less satisfactory method of regulation is by means of the stop valve, that is to say, by the engine driver actually open-

ing or closing his main steam valve, according to the output required. If this has to be done, each engine driver must be able to see the pressure to which he has to regulate, and instead of regulation taking place by an easily moved switch in the hands of one man, it has to be carried out by the action of half a dozen stop valves in the hands of perhaps as many different men. The enormous voltmeters which were made a few years ago, and I believe are still used in America in some places, having faces so large that they can be seen all over an engine room, were designed in connection with this particular method of regulation. Regulation by field resistance was perhaps then not so well understood as it is now. At any rate, there can be no doubt which method is the better on all practical grounds.

I have been speaking of efficiency of regulation so far as it affects or is affected by the gradual alterations of load, such as are represented upon the diagrams on the wall. At times of full load this is practically the only matter of importance, because at such times any sudden increase or decrease of load is small proportionately to the whole load of the station. But in a station where the load every afternoon is, say, 1,000 kilowatts, it may quite well happen that in the early hours of the morning, or during a bright forenoon, the total load is little more than a hundredth part of this. In such a case the sudden throwing on or off of a motor without proper starting resistance, or even of a photographer's arc lamp under similar conditions, may increase or decrease the actual output by 20 or 30 per cent. or even more. Such variations are most often excessively sudden and of extremely short duration, so that they are particularly difficult to regulate for, and one must only be thankful that in the nature of things they can only occur at times when a little flickering of the light is not noticeable. In stations which are able to use batteries, there is no doubt that the battery acts practically the part of a fly wheel in respect to these rapid variations.

I now come to my last section, namely, that of economy of working a station, still looking at the matter, of course, from the point of view of the mechanical engineer. Economy of working covers two sets of questions. There is, first, the reduc-

tion of direct cost of coal, oil, stores, water and maintenance per unit or per horse power hour; and secondly, the obtaining of the greatest possible ratio between the net useful and the gross indicated horse power. Taking these matters in their order, the question of coal naturally comes first. In an electric light station the cost of coal averages not far from four-tenths of the total working expenses. While, however, this matter is so important a one, it is yet one on which so much has been said and written that it need not detain us long. There is no one type of boiler —at least, I do not know of any—which is better than all others under the conditions of an electric light station. Indeed, I would go so far as to say that from the point of view of economy there are five or six types of boiler which, with equally good stoking, will yield equally good and in every respect first-class results.

One thing which tells very much against the economy of an electric light station in respect to its evaporation per pound of fuel, is the extent to which, during all day in many places, fires must be kept alight, and pressures up in boilers capable of giving eight or ten times the actual output, and the extent to which in all stations fuel must be expended in getting boilers ready for the heavy load which comes on only once in 24 hours. These stand-by losses are not, as one would think, losses which get less as the output of the station increases, for the proportion which the stand-by boilers will bear to the actually used boilers at any moment will be practically constant, whatever the output of the station. I have found that the total stand-by losses can be reduced in some cases to under 8 per cent. of the total fuel. Below this I have not yet succeeded in coming, although I hope a lower ratio will be obtained in the future. I am afraid it is often considerably more than 10 per cent.

I believe that a far larger proportion of fuel waste in a station occurs through causes beyond the boilers than in the boilers themselves, that is to say, it is much more easy to get a good evaporation per pound of coal than a small consumption of water per indicated horse power. Of the causes which contribute to this, the first is undoubtedly the very great loss which occurs by condensation in the steam pipes and by leakage through drains

and other valves before ever the steam gets to the engines at all. This is probably much greater in proportion in an electric light station than in most other places, because the requirements of security, which I have already discussed, necessitate the use of a very elaborate system of steam pipes, much more elaborate than is found in any other works. Not only does this system of steam pipes present a large radiating surface, and in various ways offer particular facilities for cooling the steam, but also the absolute necessity of keeping the pipes clear of water at all times, so that any section of them can be used safely at a moment's notice, involves the use of drain pipes and cocks. These necessary adjuncts to a system of steam pipes, along with the apparatus which we call by courtesy steam traps, and use in connection with them, are a continual source of waste and annoyance to the engineer. The drain cocks get left open, or if they are shut they leak themselves open, the steam traps require more looking after than the whole of the rest of the machinery put together, and altogether the whole system of pipes, drains and traps probably cost us in many cases 10 per cent. of our whole fuel. In some experiments which I recently had made I have found the proportion to be 8.8 per cent., and I have reason to think that it is often far greater. Apart from this matter I know no reason why any engine in an electric light station should not work every day and every hour at exactly the economy which the same engine reached on special trials made at the same load under the strict conditions of a specification. I have, in fact, on many occasions compared the running of engines in this way with the running of the same machines at the same loads on their original tests. When I have subtracted the amount of steam used for the feed pumps, the difference was never greater than I have found to be covered by the condensation losses which I have mentioned. Of course it is necessary to remember that engines working in a central station at very varying powers cannot be expected to give average results equal to those obtained at full power on trial. The results should be compared on the basis of tests at corresponding powers, and in order that

this comparison may be carried out, I always have a series of graduated tests made before taking over machinery.

After the cost of coal comes the cost of oil, water and stores. These three items together cost in the average about one-fifth as much as the cost of the coal alone. There is not much which can be said in circumstances like the present on the question of economy in this part of the expenditure, which really depends altogether on the good management of the engineer in charge. Such matters as the filtering and re-using of lubricants, and as the prevention of all needless waste of water, are obviously to be considered. It is, perhaps, generally sufficient for the engineer to find out the very lowest cost under these heads obtained in any other station, and to believe he can do a little better if only he tries hard enough. Cost of repairs and maintenance, still more than that of oil and stores, is one about which little can be formulated. There is no one type of engine or dynamo, so far as I know, that can be definitely said to be ahead of all others in respect of the fewness of its repairs. There are several types of machines which in the hands of careful engineers certainly give very good results, although it is somewhat difficult to compare the published figures under this head, on account of the number of electrical stations which are still wholly or partially in the hands of their contractors, and in which, therefore, the full amount of repairs does not appear in the accounts.

I have mentioned that the economy of a station depends on two sets of considerations, namely, the actual quantity of fuel, stores, &c., per indicated horse-power hour; and, secondly, on the actual relation between the useful power and the gross power. It is obvious that it would be of no use to save 10 per cent. on the coal per indicated horse-power hour if that saving involved the use of machinery whose mechanical efficiency was 10 per cent. lower than it would otherwise be. This brings us face to face with the question which is always one of the most important to the engineer, namely, when he has got his horse power in his engine as cheaply as possible, how to arrange so as to utilize the largest possible proportion of the horse power thus obtained. The utilized horse power is represented, in the case of an elec-

tric-light station, by the energy passing through consumers' meters and paid for by them. Between this useful horse power and the indicated horse power of the engine there comes, first, the loss in the engine itself; then the loss between the engine and dynamo; thirdly, the loss in the dynamo itself, and, fourthly, all the losses between the terminals of the dynamo and the consumer's meter. As to loss in the engine itself, this does not appear to vary very much in different types of engine of first-class make. Approximately, it may be taken as about 10 per cent., sometimes a little less, of the full power of the engine, and as remaining very nearly constant at all powers as long as the speed is constant. Thus the mechanical efficiency of an engine which is 90 per cent. at full load will be only about 80 per cent. at half load. The efficiency of a dynamo at full load is much higher, it may be as much as 95 per cent., and may be still 90 per cent., far below half load. Thus the ratio of electrical to indicated horse power of a first class steam engine and dynamo may be 85 per cent. at full load, and something like 76 per cent. at half load. This assumes that the engine drives the dynamo direct, and that there is therefore no loss in transmission between them. I must confess that it remains a complete puzzle to me, a puzzle of which I have never found any solution, why engineers should wish, in addition to these practically unavoidable losses, to spend certainly 5 per cent., and on the average probably much more, of the whole of the power in their station in moving ropes or leather belts which have nothing whatever to do with the ultimate result, and which take up an enormous amount of space. The true solution of the matter really appears to me beyond all doubt to be either the use of quick-running engines, as is so common in this country, or of slow-running dynamos, as is so commonly the case on the Continent. I am, of course, here speaking of driving by steam engines, not by gas engines, in which latter case a number of other considerations come in.

As most steam engines take practically the same power per revolution to drive themselves, it is obvious that at low powers an increase of mechanical efficiency will be obtained by running at a less speed than that used for full power. This, of course,

requires a specially arranged adjustable governor and a wide range of control in the field regulation. But I think it is worth doing in many cases.

The losses between the dynamo terminals and the consumers' lamps are, in a low-tension system, simply the losses in the leads. In a high-tension system they cover the losses, in general very much smaller, in the leads, but also those in the transformers as well. This is not a place nor a time for entering on a discussion of the relative merits of high and low tension systems—a matter which to anyone who is not a partisan of one or the other could certainly not be adequately discussed in any time short of the whole of that which has been occupied by this paper. Moreover, I need hardly point out that the choice of a system cannot possibly rest upon any one single point, such as the relative magnitude of the losses for a single cause. At present, so far as the figures to which I have access go, it may be interesting to you if I mention that in the case of a low-tension system where the maximum proportion of loss in the feeders is allowed to reach 20 per cent. or thereabouts, the actual average loss of energy throughout the whole year amounts to about 10 per cent. This is, of course, entirely due to ohmic resistance of the feeders themselves and of the network. I have no statistics by which I can compare feeder losses only in an alternating-current system with the figure I have just mentioned, but so far as those figures go to which I have access, apparently the total losses both in mains and transformers in a high-tension system are not less than 25 per cent. of the energy generated. It is probable, indeed I think it is certain, that this figure will be very considerably reduced in the cases where banked transformers are employed with low-tension distributing mains. I hardly think, however, that it can be expected that the total losses will ever be so low as with a low-tension system.

I have already alluded to the extent to which the economy of a station is affected by the average load at which its units are run. To attain the greatest economy it is obviously advisable that the machines actually at work at any moment should all be as nearly as possible fully loaded. The extent to which this

can be actually reached in practice varies greatly. In one case that came under my notice the average load during 24 hours was 78 per cent. of the load which would have been reached if the same machines had run the same number of hours each at its full power. In another case the percentage was 58. What this particular percentage may be always depends on the exact relations between the load and the size of the units, and therefore in any station may vary very greatly at different times of the year. It is, however, not at all as easy as has often been thought to make the load factor, to use the expression in the sense of the above ratio, very high. I find it difficult enough in many cases, even with the use of comparatively numerous units, to keep it up to an average even of 65 per cent. Of course it has to be remembered that whether or not it is economical, the station engineer has to make certain that he has sufficient plant running to take up the work as it comes on. It is much the less of two evils that he should have every engine running every day half an-hour before it is wanted, than that once a year he should be just too late with one or two of his machines. When the system of transformer sub-stations comes to be worked out practically a little more than it is at present, I believe that this load-factor problem will be found to be there even more troublesome than it is in the engine room, and that the capacity of the transformers which the engineer will consider it necessary to keep on circuit in order that he may be always safe against breakdown of any of them, will be considerably greater than has been often assumed.

I would only, in conclusion, mention one other matter. The mechanical engineer may be supposed to be of all others one who will most naturally take to the making of measurements of all kinds. Any passion which he may have for this has ample scope in connection with electrical work. Electrical measurements, at least with continuous currents, can be made more easily, with simpler apparatus, and more accurately, than any other such measurements. I do not doubt that this fact has helped very much in the extremely rapid progress made during the last few years in matters electrical. Apart altogether from the more re-

fined measurements which should be made from time to time for special purposes, there is no difficulty in getting continuous measurements, self-registered, with an average error of not more than 1 per cent., of the current sent out from each dynamo in a station. The average station volts can also be easily obtained with fair accuracy, and the amount of electrical energy developed at the dynamo terminals can thus be continuously measured as no other similar amount ever has been measured. This amount has to be compared with the amount of energy actually paid for, that is, with the sum of the readings on the meters of consumers.

Some of the electrical companies have published with their annual reports detailed electrical balance sheets which show how very small the unaccounted-for quantity now is. In the case of the Westminster Electric Supply Corporation it was over 5 per cent. in 1891, 3.16 per cent. in 1892, and was still further reduced to 1.28 per cent. in 1893.

Unfortunately, with alternating currents measurements are very much more difficult and troublesome, and consciously or unconsciously, I think this fact has, to a certain extent, hindered their adoption. There are now, however, alternate-current watt-meters practically free from error due to circuit induction, and capable of giving results with quite sufficient accuracy under the actual conditions of station practice. I believe that very great improvements in the economy of alternate-current working will date in every case from the time when the station commences to make accurate determinations—based on something different from the engine driver's hourly entries in a log—of the true energy generated and the way in which it has been expended.

Regarding Professor Kennedy's statement that, in electrical traction, the total average efficiency does not exceed 35 per cent., Mr. C. S. DuRiche Prellier, writing to "Engineering," says:

Surely this is either a mere conjecture, or Professor Kennedy means that 35 per cent. is the total *loss*, so that the average efficiency is 65 per cent., which is nearer the mark.

There are, within my own experience, at least eight electrical lines worked by spur-geared motors, where the total average efficiency ranges from 65 to 72 per cent. In one of these cases

—a steep-grade adhesion line with 8 per cent. grades—the *guaranteed* efficiency is 60, the actual efficiency 65 per cent.

No less astonishing are the remarks in which Professor Kennedy pits cable traction and compressed gas motors against electrical traction, to the advantage of the former, an assertion which is hardly worthy of the *fin siècle*. As regards cable traction, I hope before long to show conclusively that even this can be more economically worked by electric motive power than by steam direct. And with reference to gas and oil or compressed gas motors, Professor Kennedy surely does not seriously contend that even for ordinary tramway work they possess anything like the general advantages, or can develop anything like the tractive power of electric motors fed from a central station? On the line with 8 per cent. grades already mentioned, as well as on another quite recently opened line on the Continent, with a continuous 7 per cent. gradient, the electric motor cars, at an adhesion of one-eighth, run up the incline at a uniform speed of eight miles per hour, a performance which, at equal total car-load, is out of the question either with steam, gas, oil, or compressed gas motors.

Professor Kennedy's contention would have some foundation if, instead of compressed gas motors, he had shown a predilection for compressed air motors, because these motors undoubtedly do provide a perfectly smooth and noiseless means of mechanical traction, free from smoke, odors or conductors of any kind. Successful examples may be found on the Nogent tramways near Paris, and in an even more interesting and attractive form on the Berne tramway in Switzerland, and because the system is admirably adapted as the best substitute for electricity in provincial towns.

I do not know what results Professor Kennedy expects from the compressed gas motors of whose "possibilities" he speaks so confidently, but the following practical examples conclusively show, I venture to think, the comparative weights and powers of vaporized oil (Conelly), compressed air, and electric motors, the weight including 36 passengers:

	Weight. tons.	*Power. H.P.*	*Grade. per cent.*
Conelly vaporized oil motor plus 1 car,	10	15	2.5
Berne compressed air motor car, . .	9	30	5
Florence and Fiesole electric motor car,	8	40	8

Mr. Frank B. Lea, in the same paper comments as follows:

We are told that the total efficiency from indicated horse power in the station steam engines to the driving axles on the line is only 35 per cent. in the case of an electric railway, as compared with 80 per cent. in ordinary steam working. There is no need just now to dispute the latter efficiency; it may be assumed correct; but Professor Kennedy will be nearer the truth if he doubles the ratio in the former case. Otherwise, I cannot credit the 88 per cent. ratio between station indicated horse power and electric horse power on the line which was recently stated by Mr. Thomas Parker before the Institution of Civil Engineers to represent actual working results on the Liverpool Overhead Railway. Take an even less favorably situated and operated line, the Mont Salève electric rack railway. The working results upon this are as follows:

Loss per cent. in generators, ...	10
Loss per cent. in line, ...	12
Loss per cent. in motors and gearing, . . .	26

and the mean efficiency is stated to be 52 per cent.—a low percentage, due to the unfavorable arrangement of the motors and gearing. This latter line, be it remembered, is one where gearing is essential, but on ordinary railways the idea of introducing gears at all between motors and driving axles would be scouted as absurd by any one who has studied the subject at all. Yet Professor Kennedy allows for an efficiency (or rather inefficiency) of 75 per cent. in this item alone, which is absolutely non-existent in a well-designed road. Then he deducts 10 per cent. from the efficiency for loss of energy due to deadweight of electric locomotive, but makes no allowance whatever apparently for a similar loss in the case of steam locomotives. Yet the respective deadweights are almost as 1 to 6 or even 10. For the figures

which he gives in order to arrive at the total efficiency the following may, I think, with far more accuracy be considered reliable:

	Per cent.
Mechanical efficiency of engine (preferably direct coupled),	85
Efficiency of dynamo,	95
Efficiency of line,	95
Efficiency of motors,	90
Total,	69

In all this, however, it must not be forgotten that a test of the mechanical efficiency of the steam motor is not the furthest step we can take backwards, for the ratio between fuel efficiencies at full or even average loads of the stationary and locomotive steam motors respectively will be at least 5 to 3, if not more. Although, therefore, the total efficiency as given above of an electric line be only 69 per cent. of the engine power, whilst that of a steam line is assumed at 80 per cent., yet the final ratio, based on the coal bill, will be, say, 34.5 and 24 respectively—a very material difference.

THE FRENCH TORPEDO-BOAT *CHEVALIER.*

[Translated from "Revue Industrielle."]

Now that the power and speed of our ships of war are being discussed, and comparisons made with the fleets of foreign countries, it may be interesting to show that private French builders are not behind those of other countries, and that they are studying and improving our naval *materiel.* A paper by M. de Dax, read before the Society of Civil Engineers of France, on the trials of the torpedo-boat *Chevalier* furnishes the opportunity.

This boat is from the works of Mr. Augustin Normand, who has already delivered three boats of the same type to the French Navy—the *Dragon*, the *Grenadier* and the *Lancier.* The last one, the *Chevalier*, differs slightly from the previous ones, and presents some peculiarities of interest. Her principal dimensions are:

Length between perpendiculars, feet, . .	144.33
Breadth at L.W.L., feet, ...	14.40
Depth, feet, ...	4.43
Displacement, tons, ...	117.58

The ratio of length to beam is practically 10 to 1, while the *Lancier*, her immediate predecessor, had a ratio of 9.5 to 1, being 137.76 feet long and 14.43 beam, with a displacement of 118.36 tons.

On account of the difference in dimensions, and the internal arrangements adopted in the *Chevalier*, she carries only 17 tons of coal against 22 in the *Lancier.*

One of the most interesting parts of this type of boat is the stern, the principal characteristics of which are a long overhang, the rudder arranged forward of the screws, and the rotation of the screws in the same direction, the circles described by the tips of the blades partially overlapping each other.

Since 1890, Mr. Normand has held that the rudder placed forward of the screws should have a turning power at least equal

to what it would have if placed abaft them; provided that, in the first case, the water shed by the rudder can regain its normal direction before reaching the screws.

The rudder of the *Lancier* was placed forward of the screws, and that of the *Dragon* aft. The time of turning, and the ease of handling, was the same in each case, and if anything slightly in favor of the *Lancier*. These results induced Mr. Normand to fit the same arrangement on the *Chevalier*, as it embodies certain advantages. It gets rid of a part of the keel aft; the rudder is not so much out of water when the boat is pitching; and the breaking of the column on one of its faces is less frequent at high speed.

The *Chevalier*, as before stated, is fitted with two screws, each in a different transverse plane. The result is that the shafts have been placed closer together than would have been possible with the usual arrangement, and that, as before stated, the circles described by the tips of the blades partially overlap each other. Hence, there is a reduction in the lateral projection of the propellers, thus avoiding a frequent cause of accident.

The dimensions of the screws are:

Diameter, feet, ...	5.85
Mean pitch, feet, ...	9.05

Hitherto, in vessels propelled by two screws, the screws have turned in opposite directions, one right and the other left handed. In the *Chevalier* both screws are right handed, with the same pitch, and both turn in the same direction. This arrangement has been adopted by Mr. Normand for the following reasons:

If we look at what takes place in the case of two screws turning in opposite directions, it is readily seen that the water set in motion by one of the blades, as *A* in the sketch, has an ascending motion; the blade *B*, which also moves upward, moves in a fluid having motion in the same direction as itself, and which, consequently, has a certain motion in advance of it. In the second arrangement, however, the blade *A* gives the water an upward motion, but the blade *B* tends to give it motion in the opposite direction. Consequently, the reaction is greater, and the efficiency of the screw blades greater.

Finally the cutting away of the keel near the screws facilitates the motion of the particles of water tending to replace those set in motion by the rotation of the screws.

As to the influence of the immersion of the screws and the speed upon the breaking of the column of water acted upon, Mr. Normand has recently read a paper before the Association Technique Maritime, in which he demonstrated the importance of it.

Boilers and Engines.—The boilers are of the DuTemple type, two in number, and are placed in a common boiler room. Each has 826 tubes and contains 43 square feet of grate and 1,642 of heating surface, or 3,284 square feet of heating surface for the two boilers.

The engines are of the twin screw triple-expansion type, and are provided with various appliances for reducing as far as possible the vibrations due to the inertia of the moving parts.

The boilers are provided with feed-water filters and feed-water heaters. After having tried various substances, such as cotton waste, gauze and coke, Mr. Normand finally decided to use sponge in his filter; and as now constructed it consists of three layers of sponge superposed, and placed either in the feed tank or between the air pump and the feed pump.

The feed heater is a surface heater; that is, the steam does not come in direct contact with the water, and it is placed between the feed pump discharge and the boiler. In Torpedo Boats *Nos. 126* and *129*, with compound engines, the steam for heating the feed was taken by a special valve from the middle of the stroke of the low-pressure cylinder, but in the boats with triple-expansion engines it is taken from the low-pressure steam chest; in other words, the steam used for heating the feed water has already given up two-thirds of its work before being used in the heater. The efficiency of these heaters has been greatly increased since 1892, and during a test at low power the temperature of the feed has been raised from 187 to 203 degrees and at full power from 194 to 241 degrees. According to Mr. Normand, the trials of a number of torpedo boats seem to show that at a speed of from 10 to 15 knots there is an economy of 20 per cent. due to the use of the heaters.

The special accessories on the engines are the automatic traps and the cylinder relief valves. Each engine has three traps, one for the high-pressure steam chest, one for the jackets, and the other for the feed-water heater. The special relief valves are for the purpose of preventing excessive compression; when the compression exceeds the pressure in the steam chest, they open and allow the steam to return to the chest. They have been in use for some time and have given good results.

Mr. Normand has given special attention to the question of vibration due to the inertia of the reciprocating parts, and has reached the following conclusion:

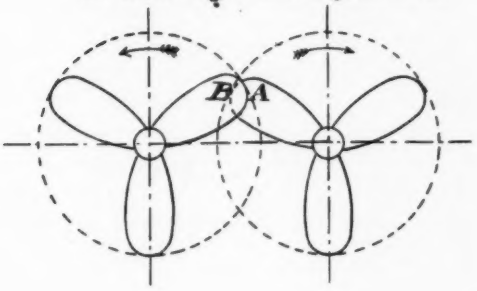
With three cranks at 120 degrees apart, or four cranks at 90 degrees, and the same weight of reciprocating parts for each cylinder, the sum of the moments of inertia in the direction of the axes of the cylinders is *nil* for all angular positions of the cranks, whatever the ratio of the length of the connecting rod to crank may be. Upon this theory, the three pistons of the *Chevalier* have been made almost exactly the same weight; on account of the air, bilge and feed pumps being worked from the I. P. piston rod, the latter piston is about 6.6 pounds lighter than the others.

The arm of the couple producing vibration may be considered as proportional to the distance between the axes of the extreme cylinders, and in order to reduce this distance to a minimum, the valves have been placed at the sides of the cylinders.

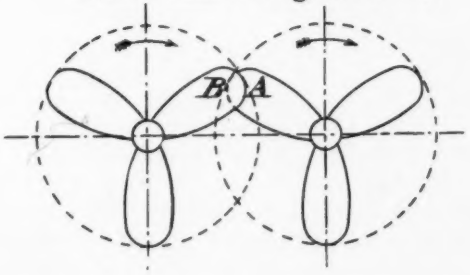
In addition, the engines have been made rigid in a longitudinal plane by means of diagonal bracing, and special braces also tie them to the deck and to the beams, so that the strains which are transmitted are taken up by these parts. Experience has shown that this method of tying the engine to the deck and the beams gives the best results, and the official report says that the vibrations were almost imperceptible.

Armament.—Being a boat intended by her speed to escape pursuit by an armor clad or even a large cruiser, she has only to defend herself against boats of her own class, and so a light defensive armament suffices. She carries, in fact, only two 37-mm. rapid-fire guns. For offensive purposes she has two 18-inch torpedo tubes mounted on carriages.

Ordinary Arrangement.



New Arrangement.



Speed.—There were two trials, the first one for economy at a speed of 10 knots, and the second at full speed.

To obtain the speed of 10 knots but one engine was used, the starboard one, with only one boiler, the grate surface of which had been reduced from 43 to 31 square feet. The following results were obtained:

Revolutions of port screw, disconnected,	117.
Revolutions of starboard screw,	149.3
Speed, knots,	10.326
Steam pressure, pounds,	142.
Coal per hour, pounds,	174.
Coal per hour per square foot of grate, pounds,	5.61
Coal per knot, reduced to a speed of 10 knots, pounds,	15.4

The full speed trial gave the following results:

Revolutions per minute,	360.66
Speed, knots,	27.22
Steam pressure, pounds,	211.
I.H.P., estimated,	2,700.
Coal per hour, pounds,	5,378.
Coal per hour per square foot of grate, pounds,	62.5
Coal per knot, pounds,	197.6
Coal per I.H.P. (approximate), pounds,	1.99

During this trial, it was observed that the boat had a list of 2° 10′ to port due to the arrangement of the screws, and that the port screw being thus deeper immersed was more efficient. The engines seemed capable of working off more steam, but the boilers could not furnish it, and the trial was stopped on account of the hard work on the firemen, each of whom had to fire about 44 pounds of coal per minute.

TRIAL DATA OF STEAMSHIPS.

By J. M. Lincoln, Esq., Associate.

The accompanying tables, giving the principal dimensions and the trial speed and power of steamships, have been in my possession for some time, and, as I have found the data therein contained of much service in sundry calculations of projected work, I submit them for the use of other members of the Society.

	Sabyholin.	*Ceres.*	*Angjulli.*	
Moulded dimensions, feet	72.46 × 16.56 × 9.5	228 × 30 × 21.67	38 × 8 91 × 4.29	64.
Length on water line, feet	89.	223.	35.	
Mean trial draft, feet	5.5	13.58	3.5 2.0	
Depth of keel, inches	3.	8.	.....	
Moulded draft, feet	5.17	12.91	2.175	
Displacement, tons	95.	1,670.	7.7	
Displacement tons × 35 cubic feet ÷ ⊕ × L.W.L. = δ	0.56	0.72	0.61	
Area ⊕ square feet	67.4	363	12.53	
Entrance angle mean, degrees and minutes	12	16	12	
Total surface, square feet	1,650	8,920	333	
Propeller tips, in or out of water	In.	.....	.....	
Area of propeller disc, { Total	23	.....	.....	
Area of propeller disc, { Immersed	23	.....	7.068	
Immersed area, square feet (propeller) ÷ immersed bottom surface in units of 100 square feet	1.393	.....	2.122	
Knots.	I.H.P.	I.H.P.	I.H.P.	
5.	12	185	.....	
6.	22	228	.....	
7.	33	283	.....	
8.	50	.....	7.75 — 24.5 8 — 29	
8.5	62	390	.....	
9.	75	440	.....	
9.5	92	490	.....	
10.	115	546	.....	
10.5	144	625	.....	
11.	184	718	.....	
11.5	232	850	.....	
12.	288	1,014	.....	
12.5	.....	.....	.....	
13.	.....	.....	.....	
13.5	.....	.....	.....	
14.	.....	.....	.....	
14.5	.....	.....	.....	
15.	.....	.....	.....	
15.5	.....	.....	.....	
16.	.....	.....	.....	
16.5	.....	.....	.....	
17.	.....	.....	.....	
17.5	.....	.....	.....	
18.	.....	.....	.....	
19.	.....	.....	.....	
20.	.....	.....	.....	
$D^{\frac{2}{3}} \times S^3 \div$ I.H.P.	149	239	68	
Highest speed and powers. I.H.P.	190	1,014	29	
Highest speed and powers. Speed.	11.08	12.	8.	

TRIAL DATA OF STEAMSHIPS.

Ophelia.	*Raynhild.*	Torpedo-Boat *Seid.*	Torpedo-Boat, Thornycroft.	*Shakspeare.*	*Shakspeare.*	*Umbria.*	Twin-sc low-draft *Co*
64.12×15.58×10.15	111.7 × 21.5 × 10.21	.....	.....	272 × 35.79	272 × 35.79	500 × 57	70 ×
64.67	108.	101.	83.75	270.	270.	500.	7
6.58	7.76	.....	.....	9.79	9.79	22.33	
4.	4.	.....	.....	9.5	9.5	12.	.....
6.25	7.45	2.5	2.5	9.0	19.43	21.33	
62.2	141.	42.1	32.44	1,666.	4,087.	9,860.	
0.52	0.5	0.64	0.65	0.745	0.805	0.6902	
64.05	91.87	25.	20.7	290.5	659.	1,000.	
12 15	9 30	8 10	8 12	12 30	17 16	8 36	8
1,090	2,190	950	848	11,491	18,122	35,061	56
.....	.....	.....	.....	Out.	In.	Out.	O
.....	.....	.....	.....	168.7	168.7	471.	
42.	52.8	.....	.....	131.2	168.7	435.	
3.84	2.41	.....	.....	1.21	0.931	1.25	
I.H.P	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H
.....	.....	.....	.....	.....	.....	.....	
.....	.....	.....	.....	.....	.....	.....	
.....	.....	.....	.....	230	388	860	
70	.....	.....	12	356	542	1,050	
84	.....	.....	26	430	654	1,150	
104	.....	.....	31	515	756	1,280	
134	.....	.....	37	600	891	1,430	
172	.....	.....	45	705	1,043	1,600	
.....	.....	.....	55	810	.....	1,750	
.....	.....	.....	66	930	.....	1,950	.....
.....	275	79	77	1,060	.....	2,160	.....
.....	370	100	92	1,200	.....	2,400	.....
.....	12.12 408	125	110	.....	.....	2,680	.....
.....	.....	145	131	.....	.....	3,040	.....
.....	.....	167	154	.....	.....	3,400	.....
.....	.....	190	182	.....	.....	3,800	.....
.....	.....	210	210	.....	.....	4,300	.....
.....	.....	232	240	.....	.....	4,800	.....
.....	.....	253	270	.....	.....	5,420	.....
.....	.....	275	294	.....	.....	6,050	.....
.....	.....	.....	314	.....	.....	6,780	.....
.....	.....	319	.....	.....	.....	7,570	.....
.....	.....	339	.....	.....	.....	8,480	.....
.....	.....	362	365.4	.....	.....	9,420	.....
.....	.....	405	395.4	.....	.....	11,542	.....
.....	.....	446	425.5	.....	.....	13,920	.....
93.22	118	221	191	201	246	263.8	
158.5	410 6	455	425	1,005	836	14,321	6
9.8	12.15	10.25	10.	11.3	9.3	20.18	10

TRIAL DATA OF STEAMSHIPS.

	Angjulli.	*Ophelia.*	*Raynhild.*	Torpedo-Boat *Seid.*	Torpedo-Boat, Thornycroft.	*Shakspeare.*	*Shakspeare.*
1.67	38 × 8 91 × 4.29	64.12 × 15.58 × 10.15	111.7 × 21.5 × 10.21	.....	.....	272 × 35.79	272 × 35.79
	35.	64.67	108.	101.	83.75	270.	270.
	3.5 2.0	6.58	7.76	.....	.....	9.79	9.79
	.....	4.	4.	.....	.....	9.5	9.5
	2.175	6.25	7.45	2.5	2.5	9.0	19.43
	7.7	62.2	141.	42.1	32.44	1,666.	4,087.
	0.61	0.52	0.5	0.64	0.65	0.745	0.805
	12.53	64.03	91.87	25.	20.7	290.5	659.
	12	12 15	9 30	8 10	8 12	12 30	17 16
	333	1,090	2,190	950	848	11,491	18,122
	.....	.....	.....	.....	.....	Out.	In.
	7.068	42.	52.8	.....	.....	168.7	168.7
	2.122	3.84	2.41	.....	.....	1.21	0.931
	I.H.P.	I.H.P	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	230	388
	7.75 — 24.5 8 — 29	70	.....	.....	12	356	542
	.....	84	.....	.....	26	430	654
	.....	104	.....	.....	31	515	756
	.....	134	.....	.....	37	600	891
	.....	172	.....	.....	45	705	1,043
	.....	.....	.....	.....	55	810	.....
	.....	.....	.....	.....	66	930	.....
	.....	.....	275	79	77	1,060	.....
	.....	.....	370	100	92	1,200	.....
	.....	.....	12.12 408	125	110	.....	.....
	.....	.....	.....	145	131	.....	.....
	.....	.....	.....	167	154	.....	.....
	.....	.....	.....	190	182	.....	.....
	.....	.....	.....	210	210	.....	.....
	.....	.....	.....	232	240	.....	.....
	.....	.....	.....	253	270	.....	.....
	.....	.....	.....	275	294	.....	.....
	.....	.....	.....	.....	314	.....	.....
	.....	.....	.....	319	.....	.....	.....
	.....	.....	.....	339	.....	.....	.....
	.....	.....	.....	362	365.4	.....	.....
	.....	.....	.....	405	395.4	.....	.....
	.....	.....	.....	446	425.5	.....	.....
	68	93.22	118	221	191	201	246
	29	158.5	410 6	455	425	1,005	836
	8.	9.8	12.15	20.85	20.	11.3	9.3

eare.	*Umbria.*	Twin-screw, shallow-draft Steamer *Congo.*	*Charles V.*	*Kronprintz.*	*Skagul.*	Twin-screw, *Urd.*	*Chelydia.*
5.79	500 × 57	70 × 10.6	323 × 33	286 × 43.4	171.4 × — × 25.8	171.4 × — × 25.9	300 × 39 83
	500.	70.	323.	286.	171.42	171.42	291.2
79	22.33	1.	14.83	23.61	9.5	9.5	17.87
5	12.	.....	7.	.....	.....	.....	10.
43	21.33	1.	14.25	.....	.....	.....	17.04
	9,860.	9.23	2,479.	5,668.	526.	526.	4,000.
805	0.6902	0.593	0.6392	0.69	0.623	0.623	0.77
	1,000.	8.0	400.	1,000.	182.	182.	623.
16	8 36	8 0	7 15	10 30	9 10	9 10	14 0
2	35,061	564	14,045	21,000	5,920	5,920	10,560
	Out.	Out.	Out.	.....	.....	.....	In.
7	471.	1.4	201.	.....	.....	.....	214.
7	435.	0.76	189.	.....	.....	.....	214.
931	1.25	0.135	1.345	.....	.....	.....	1.153
P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
...	.....	6.	105	.....	.....	.....	.....
...	.....	9.5	148	320	.....	.....	.....
	860	15.	210	480	.....	.....	.....
	1,050	23.	284	720	.....	.....	392
	1,150	28.	330	870	.....	.....	480
	1,280	34.	385	1,020	.....	.....	575
	1,430	41.5	448	1,260	.....	.....	685
	1,600	50.	520	1,500	250	.....	798
..	1,750	60.	595	1,850	300	.....	930
..	1,950	.....	688	2,200	355	.....	1,075
..	2,160	.....	800	2,590	420	.....	1,230
..	2,400	.....	910	2,980	500	.....	1,390
..	2,680	.....	1,050	3,430	596	.....	1,560
..	3,040	.....	1,208	3,870	710	786	.....
..	3,400	.....	1,380	4,300	848	962	.....
..	3,800	.....	1,578	4,770	.....	1,170	.....
..	4,300	.....	1,795	5,240	.....	.....	.....
..	4,800	.....	2,025	.....	.....	.....	.....
..	5,420	.....	.....	.....	.....	.....	.....
..	6,050	.....	.....	.....	.....	.....	.....
..	6,780	.....	.....	.....	.....	.....	.....
..	7,570	.....	.....	.....	.....	.....	.....
..	8,480	.....	.....	.....	.....	.....	.....
..	9,420	.....	.....	.....	.....	.....	.....
..	11,542	.....	.....	.....	.....	.....	.....
..	13,920	.....	.....	.....	.....	.....	.....
	263.8	85.2	242.5	183	193	161.6	313.4
	14,321	60	2,025	5,240	810.4	952	1,435
	20.18	10.49	15.	14.5	13.39	13.42	12.128

	Iris.	*Assam.*	*Elton.*
Moulded dimensions, feet and inches	300 × 45 4 × 27.1	274.6 × 34 × 26	307 × 33 × 26
Length on water line, feet and inches	300	274.6	307
Mean trial draught, feet and inches	18 1	15 5½	13 7½
Depth of keel, inches	11	9	10
Moulded draught, feet	17.17	14.7	12.79
Displacements, tons	3,290	2,338	2,311
Displacement, tons × 35 ÷ ⌓ × L.W.L. = δ	0.548	0.688	0.662
Area ⌓ section	700	433	397
Value A	13.55	85.5	103.2
Value B	20.35	14.73	15.5
Value C	29 0	103.5	100.5
Value P	137.	86.8	104.3
Angle of entrance	8° 33′	9° 46′	8° 33′
Bottom surface	6,695	5,570	6,315
Side surface	10,423	8,150	7,907
Total surface	17,118	13,720	14,222
Propeller tips in or out of water, feet and inches	2 9½ in	1 3 out	2 0 out
Area of propeller disc { Total, square feet	258	207	189
Area of propeller disc { Immersed.	258	207	169
Immersed area ÷ surface immersed in units of 100 square feet	1.51	1.46	1.19
I.H.P. 5 knots	204	.....	.....
6 knots	284	.....	.....
7 knots	392	.....	.....
8 knots	540	.....	292
8.5 knots	628	.....	340
9 knots	716	.....	410
9.5 knots	825	595	477
10 knots	944	666	578
10.5 knots	1071	803	685
11 knots	1216	940	820
11.5 knots	1380	1088	1028
12 knots	1576	1273	1250
12.5 knots	1779	1472	1569
13 knots	2020	.....	.....
13.5 knots	2285	.....	.....
14 knots	2580	.....	.....
14.5 knots	2942	.....	.....
15 knots	3300	.....	.....
15.5 knots	3742	.....	.....
16 knots	4180	.....	.....
16.5 knots	4788	.....	.....
17 knots	5280	.....	.....
17.5 knots	6040	.....	.....
18 knots	6650	.....	.....
D% × S³ ÷ I H.P.	188.	237.	234.6
Highest Speed and Powers. I.H.P.	7,556	1,336	1,333
Speed.	18.59	12 16	12.14

	Africa. 172	*Goa.*	*Hawea.* March 27, 1875.	*Hawea.* April 1, 1875.	*Taupo.*	*Africa.* 182	*Merkara.*	*Pau-T* June 23,
	315 × 33 × 26.6	297 × 33 × 26.6	215 × 27 × 14.7	215 × 27 × 14.7	215 × 27.4 × 14.7	260 × 33 × 26	369 × 37 × 29.6	225 × 34
	315	293	215	215	215	257	369	225
	14 1	14 8	11 5	13 0	11 5½	15 0	16 3	8 5
	10	10	7½	7½	7½	9	11	8
	13.25	13 88	10.83	12.47	10.81	14.25	15.33	8.3
	2,530	2,462	1,088	1,280	1,088	1,970	3,940	1,15
	0.694	0 558	0 656	0.668	0.656	0.629	0.712	0.67
	405	425	270	312	270	426	525	26
	96.4	94.2	74.0	71.5	78.0	98.1	106.3	72.
	15.28	15.36	12.4	12.6	12.4	14 95	17.12	16.
	122.2	108.6	67.0	72.0	67 0	63.7	156.4	79.
	97.6	95.4	75.0	72.6	75.0	99.2	107.7	74.
	9° 0′	9° 16′	9° 31′	10° 0′	9° 31′	8° 38′	9° 9′	12° 23
	6,680	6,230	3,497	3,616	3,497	4,822	8,796	4,86
	8,411	8,284	4,700	5,374	4,700	7,470	11,400	3.79
	15,091	14,514	8,197	8,790	8,197	12,272	20,396	8,66
	0 7 out	0 9 out	0 7½ out	0 10½ in	0 8 out	0 3 out	0 4 in	3 2½
	189	189	113	118	113	189	214	11
	186	185	111	113	116	188	214	8
	1.23	1.27	1.36	1.26	1.34	1.53	1.05	1.0
	.....	.....	95	.....	102	106	.....	9
	.....	.....	132	.....	145	160	280	15
	.....	.....	180	.....	203	233	384	23
	314	345	243	235	276	325	514	33
	367	400	290	280	323	380	595	40
	429	466	326	311	375	442	677	48
	499	535	381	395	438	514	780	58
	580	625	438	472	510	592	900	67
	682	717	516	565	596	684	1034	81
	802	844	612	681	700	790	1194	95
	948	958	785	820	815	908	1365	113
	1125	1126	912	992	946	1040	1559	135
	1340	1302	1140	1214	1100	1200	1770	162
	1591	.....	.....	.....	.....	1400	1990	.....
	.....	.....	.....	.....	.....	1000	2240	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	.....	.....	.....	.....	.....	.....	.....	.....
	269.6	281.3	2c6.6	230.0	188.1	246.4	281.3	143
	1,319	1,103	854	855	1,083	1,414	1,908	1,257
	12.42	11.94	11.86	11.86	12.44	13.04	12.91	11.79

(*a.*) To outside

TRIAL DATA OF STEAMSHIPS.

Merkara.	*Pau-Tah.* June 23, 1875.	*Pau-Tah.* July 1, 1875.	*Fung-Shun.*	*Wakatipu.*	*General Pel.*	*Chanda.*	*German.*	*M* May
369 × 37 × 29.6	225 × 34 × 21	225 × 34 × 21	225 × 34 × 21	290 × 33 × 25	255 × 31.6 × 25.9	315 × 33 × —	350 × 39 × 31	380 ×
369	225	223	225	290	255	315	350	1
16 3	8 5	12 4	11 8½	14 10¾	13 0	14 9¼	18 8¾	
11	1¼	1¼	1¼	1¾	8½	10	9½	
15.33	8.31	12.23	11.6	14.75	12 29	13.94	17.94	
3,940	1,156	1,830	1,720	2,740	1,688	2,695	4,540	3
0.712	0.676	0.719	0.713	0.719	0.669	0.701	0 703	0
525	266	399	378	460	346	428	646	
106.3	72.4	64.5	65.7	81.5	84.2	94.6	104.0	
17.18	16.0	16.3	16.3	18.6	18 07	15.35	18 0	
156.4	79.2	96.0	93.6	127.0	86.6	125 8	142.0	
107.7	74.6	66.5	67.7	83.0	85.4	95.8	105.5	
9° 9′	12° 23′	14° 11′	13° 36′	10° 51′	9° 29′	9° 13′	9° 49′	
8,796	4,867	5,232	5,192	6,501	4,806	6,766	8,860	
11,400	3.796	5.600	5,312	8,642	6,332	8,849	12,670	1
20,396	8,663	10,832	10,504	15,143	11,158	15,615	21,530	1
o 4 in	3 2½ out	o 2½ in	o 4½ in	o ½ out	o 4 out	o 4 out	o 7 in	2
214	113	113	113	177	143	184	240	
214	89	113	113	177	142	188	240	
1.05	1.03	1.04	1.08	1.17	1.27	1.20	1.11	
.....	97	100	.....	114	.....	130	195	.....
280	153	153	.....	173	.....	188	268	.....
384	230	227	255	261	.....	264	364	
514	335	326	354	379	284	364	485	
595	403	387	405	450	335	428	564	
677	482	473	485	531	395	504	652	
780	580	570	555	625	465	586	756	
900	678	690	667	736	540	692	876	
1034	814	835	790	852	624	866	1025	
1194	950	1007	941	1026	729	940	1182	
1365	1131	1222	1158	1220	849	1093	1376	
1559	1350	1480	1454	1444	996	1270	1595	
1770	1628	1790	1907	1712	1160	1478	1850	
1990	.....	.....	.....	.....	1362	.....	2135	
2240	.....	.....	.....	.....	.....	.....	2465	
.....	.....	.....	.....	.....	.....	.....	2850	
.....	.....	.....	.....	.....	.....	.....	3293	
.....	.....	.....	.....	.....	.....	.....	.....	
.....	.....	.....	.....	.....	.....	.....	.....	
.....	.....	.....	.....	.....	.....	.....	.....	
.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
281.3	143 2	183.6	183.5	240 5	229.1	262.4	265.	
1,968	1,257	1,275	1,243	1,328	1,322	1,314	2,760	
12.91	11.79	11.61	11.67	11.76	12.88	12.12	13.87	

(*a.*) To outside of planking.

Manora. May 10, 1878.	*Manora.* May 29, 1879.	*Glasgow.*	*Colaba.*	*Byculla.*	*Shunlee.* July 25, 1878.	*Shunlee.* July 31, 1878.	*Pretoria.*	*Ch* Dec. 1
380 × 38 × 30.6	380 × 38 × 30	205 × 30 × 18	264 × 32 × 28	264 × 32 × 28	230 × 35 × 28.9	230 × 35 × 28.9	350 × 40 × 32.6	285 × 3
380	380	.....	264	264	230	230	350	2
14 4½	16 5¼	13 2 (*a*.)	11 7½	11 7½	9 0½	13 5	18 11½	13
11	11	.....	9½	9½	11	1½	11	
13.46	15.52	12.2	10.83	10.83	8.93	13.32	18 04	12
3,531	4,210	1,066	1,778	1,778	1,300	2,095	4,588	2,3
0 702	0.717	0.603	0.746	0.746	0.67	0.71	0.689	0.7
463	541	302	316	316	295	449	666	
113.1	107.6	71.5	67.1	67.1	75.8	66.7	109 0	
17.21	17.43	12.58	14.6	14.6	16 48	16.85	18.46	
153.8	164.8	52.0	129.8	129.8	78.4	96.6	132.0	1
114.4	109.0	72.6	68.6	68.6	77.6	68 8	110 5	
8° 39′	9° 12′	9° 49′	12° 17′	12° 17′	12° 16′	14° 11′	9° 37′	11°
9,190	9,500	3,058	5,749	5,749	5,082	5,503	8,900	6,
10,280	11,880	4,812	5,783	5,783	4,181	6,239	12,740	7
19,470	21,380	7,868	11,532	11,532	9,263	11,742	21,640	13
2 6 out	0 5 out	1 4½ out	0 10½ out	1 1½ out	2 9 out	0 11 in	0 9½ in	1
240	240	133	177	177	123	123	240	
219	239	125	173	170	103	123	240	
1.12	1.12	1.59	1.5	1 48	1.11	1.05	1.11	
.....	.....	89	78	95	83	91	164	
.....	250	131	131	139	127	136	239	
386	360	186	188	194	185	200	338	
516	500	259	282	260	263	290	470	
595	590	302	336	312	312	355	549	
678	683	350	388	368	372	434	639	
780	795	405	453	434	450	531	750	
880	914	468	525	512	548	660	881	
1003	1050	543	614	609	675	825	1038	
1140	1215	638	713	719	833	1021	1212	
1300	1410	762	834	848	1037	1270	1427	1
1490	1630	928	.....	.....	1300	1613	1662	1
1688	1870	1156	.....	.....	1642	2065	1941	1
1930	2150	1500	.....	.....	.....	.....	2265	.....
2182	2455	.....	.....	.....	.....	.....	2655	.....
2490	2800	.....	.....	.....	.....	.....	3120	...
2804	3185	.....	.....	.....	.....	.....	3711	.....
3210	3650	.....	.....	.....	.....	.....	.....	.....
3608	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....
245.	242.6	169 5	274.8	275.5	148.0	183.1	251.	
3.195	3.185	1,246	669	649	1,498	1,488	2.804	
14.88	14 43	12.65	10.78	10.68	12.3	11.85	13.66	

...ew.	*Colaba.*	*Byculla.*	*Shunlee.* July 25, 1878.	*Shunlee.* July 31, 1878.	*Pretoria.*	*Chilka.* Dec. 10, 1878.	*Chilka.* Dec. 25, 1878.
...0 × 18	264 × 32 × 28	264 × 32 × 28	230 × 35 × 28.9	230 × 35 × 28.9	350 × 40 × 32.6	285 × 35 × 26.6	285 × 35 × 26.6
...	264	264	230	230	350	285	285
... 2 (*a.*)	11 7½	11 7½	9 0½	13 5	18 11½	13 0½	20 11
...	9½	9½	1½	1½	11	10	10
...9	10.83	10.83	8.93	13.32	18 04	12.2	20.08
...56	1,778	1,778	1,300	2,095	4,588	2,330	4,197
...23	0.746	0.746	0.67	0.71	0.689	0.717	0.703
...32	316	316	295	449	666	399	674	
...·5	67.1	67.1	73.8	66.7	109 0	80.6	67.1
...58	14.6	14.6	16 48	16.85	18.46	16.35	16.8
...·0	129.8	129.8	78.4	96.6	132.0	123.8	150.9
...·6	68.6	68.6	77.6	68 8	110 5	82.2	69.1
...49′	12° 17′	12° 17′	12° 16′	14° 11′	9° 37′	11° 28′	14° 4′
...56	5,749	5,749	5,082	5,503	8,900	6,684	7,325
...12	5,783	5,783	4,181	6,239	12,740	7,032	11,610
...68	11,532	11,532	9,263	11,742	21,640	13,716	18,935
...6 out	0 10½ out	1 1½ out	2 9 out	0 11 in	0 9½ in	1 3 out	4 5 in
...3	177	177	123	123	240	201	201
...85	173	170	103	123	240	194	201
...59	1.5	1 48	1.11	1.05	1.11	1 41	1.06
...9	78	95	83	91	164	105	.....
...1	131	139	127	136	239	160	197
...6	188	194	185	200	338	231	308
...9	282	260	263	290	470	326	452
...2	336	312	312	355	549	385	541
...0	388	368	372	434	639	455	645
...5	453	434	450	531	750	541	765
...8	525	512	548	660	881	648	912
...1	614	609	675	825	1038	737	1082
...8	713	719	833	1021	1212	885	1273
...2	834	848	1037	1270	1427	1033	.....
...8	.....	.....	1300	1613	1662	1188	.....
...0	.....	.....	1642	2065	1941	1352	.....
...	.....	.....	.....	.....	2265	.....	.....
...	.....	.....	.....	.....	2655	.....	.....
...	.....	.....	.....	.....	3120	.....	.....
...	.....	.....	.....	.....	3711	.....	.....
...	.....	.....	.....	.....	.....	.....	.....
...	.....	.....	.....	.....	.....	.....	.....
...	.....	.....	.....	.....	.....	.....	.....
...	.....	.....	.....	.....	.....	.....	.....
...	.....	.....	.....	.....	.....	.....	.....
5	274.8	275.5	148.0	183.1	251.	255.8	282.2
...240	669	649	1,498	1,488	2.804	1,173	1,034
...2.05	10.78	10.68	12.3	11.85	13.66	11.96	10.39

	Scindia.	*Sirdhana.*	*Rotomahana.* July 11, 1879.	*Rotomahana.* July 14, 1879.	*Nemesis.*	Twin-screw Steamer.	Awning-decked Steamer.	Knots.
	310 × 39 × 27.6	310 × 39 × 27.6	285 × 35 × 25	285 × 35 × 25	340 × 41 ×	.....	.....	.....
	310	310	285	285	.....	118 6	232 6	.....
	15 8	18 8¼	15 0½	15 7¾	15 11	6 8¼	12 0	.....
	10	10	1⅜	1⅜	.....	6	.....	.....
	14.83	17 89	14.93	15.55	14.3	6.18	.....	.....
	3,618	4,511	2,425	2,543	2,870	211.5	1,173	.....
	0.749	9.766	0.625	0.624	0.635	0.592	0.569	.....
	545	665	478	500	465	105.6	310	.....
	77.6	72 6	107.4	107.0	124.0	.....	.....	.....
	18.37	18.6	16.0	16.1	16 26	.....	.....	.....
	154 8	164.8	70.2	71.0	92.0	.....	.....	.....
	79.7	75.0	108.6	108.1	125.1	.....	.....	.....
	13° 19′	14° 22′	8° 28′	9° 33′	7° 29′	18° 0′	27° 15′	.....
	8,538	8,831	5,686	5,731	7,024	.....	.....	.....
	9,322	11,270	8,582	8,920	9,788	.....	.....	.....
	17,860	20,101	14,268	14,651	16,812	.....	.....	.....
	0 3 out	1 6½ in	1 5½ in	0 9½ in	1 5 out	.....	.....	.....
	227	227	165	165	231	.....	.....	.....
	226	227	165	165	223	.....	.....	.....
	1.27	1 13	1.16	1.12	1.33	.....	.....	.....
	116	119	112	121	213	32	110	5
	185	181	158	171	287	48	150	6
	283	272	218	234	380	70	215	7
	412	398	298	316	498	100	290	8
	495	490	344	363	560	118	332	8.5
	591	598	400	426	650	140	385	9
	700	755	460	485	742	170	438	9.5
	830	905	537	574	859	208	500	10
	980	1 90	620	665	993	250	570	10.5
	1152	1280	723	780	1152	305	660	11
	1345	1460	840	920	1346	365	756	11.5
	1580	1665	977	1080	1592	435	870	12
	.....	.....	1130	1250	1900	507	1010	12.5
	.....	.....	1310	1465	.....	.....	1170	13
	.....	.....	1510	1700	.....	.....	1370	13.5
	.....	.....	1765	1958	.....	.....	1580	14
	....	.....	2060	2270	.....	.....	1810	14.5
	.....	....	2470	2670	.....	.....	2050	15
	.....	.....	3025	3150	.....	.....	2290	15.5
	.....	.....	.....	.....	.....	.....	2545	16
	....	....	.....	.....	.....	.....	.....	16.5
	.....	.....	.....	.....	.....	.....	.....	17
	.....	.....	.....	.....	.....	.....	.....	17.5
	.....	.....	.....	.....	.....	.....	.....	18
	266.3	288.1	226 3	231.1	216 9	138.8	179 0	.....
	1,367	1,382	2,907	2,808	1,641	470	2,545	
	11 56	11 34	15.39	15 16	12.08	12.25	16.	

NOTES.

FURTHER TRIALS OF THE BELLEVILLE BOILERS OF THE *SHARPSHOOTER*.

The following trials of the boilers of this vessel have been carried out, in addition to those which were recorded in the last number of the JOURNAL:

1st. A trial of eight hours duration at sea, with all boilers in use under natural draft:

Steam pressure in boilers, pounds,	216.
Steam pressure at engines, pounds,	140.
Vacuum in condensers, inches,	27¼
Mean revolutions of engines,	237.9
Indicated horse power,	2,620.
Speed of ship by patent log, knots,	17.

2d. A trial of three hours duration at sea, with all boilers in use under forced draft:

Steam pressure in boilers, pounds,	183.
Steam pressure at engines, pounds,	143.
Air pressure in inches of water,	0.13
Vacuum in condensers, inches,	26.4
Mean revolutions of engines,	254.75
Indicated horse power,	3,238.
Speed by patent log, knots,	19.

3d. A dock trial of eight hours duration, using the after set of boilers only, the coal consumption being limited to 12 pounds per square foot of grate. The object of this test was to compare the results with those obtained from a similar test of the locomotive boilers of the *Spanker*. The boilers of the latter were tried just before they were taken out to be replaced by the DuTemple boiler, and the evaporation was at the rate of 9.3 pounds of water per pound of coal. The results from the test of the boilers

of the *Sharpshooter* were an evaporation of 8.7 pounds of water per pound of coal, and an expenditure of 2.83 pounds of coal per I.H.P.

4th. A trial similar to the preceding one with the forward set of boilers, the results being an evaporation of 8.25 pounds of water per pound of coal, and an expenditure of 2.6 pounds of coal per I.H.P.

5th. A similar trial with the after set, but at a lower rate of combustion, the results of which were said to be quite satisfactory.

WORKING ENGINES AT LOW POWER.

A short time ago Mr. D. Croll read a paper before the Institution of Naval Architects on "Some experiments with triple-expansion engines at reduced powers," in which he showed that in the engine under test, practically the same quantity of steam was used when developing 263 as when developing 301 I.H.P., the difference being due to the method of working. The engine was of the triple-expansion type, with cylinders 13½, 21 and 35¼ inches diameter and 21 inches stroke, and the boiler a cylindrical one, 11 feet 4 inches diameter and 9 feet 1½ inches long, containing 41 square feet of grate and 1,129 of heating surface, and designed for a working pressure of 160 pounds per square inch. The cut off on the H.P. cylinder had been made about .75 the stroke, which was a greater admission than the boiler could continuously supply steam for, so in the experiments the reduced powers were obtained by reducing the boiler pressure, throttling the steam, and by running the links up. As a result of these experiments, Mr. Croll reached the following conclusions:

1. The triple-compound engine is not only singularly inelastic but also highly sensitive to wrong adjustment, and it would, therefore, appear that it will pay well to supply steamers which have to work at reduced power with a meter for measuring the feed water in order that the most economical adjustment may be determined.

2. It appears unlikely that working triple-compound engines as double-compound will lead to satisfactory results.

He gave a table in which the feed water unaccounted for by the indicator diagrams ranged from 23 to 33 per cent. at the opening to exhaust from the L.P. cylinder, and in explanation of them says:

"These percentages have been calculated for various stages of the expansion through the three cylinders. The only explanation of the cause of these enormous losses appears to be condensation and re-evaporation in the cylinders. Supposing a piston to begin its stroke with a temperature lower than the steam entering, it is evident that the steam will first heat both piston and cover faces, and, in doing so, deposit a film of water upon them. After the steam is cut off and the pressure lowered, the deposited water will partly become steam again; but when the exhaust is opened and the pressure still further lowered, the remainder will evaporate much more rapidly, and, in doing so, cool the piston, cover and cylinder walls, so that with the following stroke the cycle is again repeated. It is worthy of remark that in a triple-compound engine the steam produced by re-evaporation in the high and intermediate-pressure cylinders may possibly be usefully employed in the low-pressure cylinder; any water on the covers and pistons of the latter, however, is, upon re-evaporation, rejected as steam to the condenser.

"It appears to me to be a most important point to note, that if we assume 30 per cent. of the total feed water deposited as water on the low-pressure piston and covers, the thickness of this water film will but amount to $\frac{1}{640}$ inch. Having, therefore, to consider films of such infinitesimal thickness, it seems worth while to enquire whether the nature of the surfaces with which the steam comes into contact may not have a considerable influence upon the amount of the condensation. As far as we know at present, the cylinder walls are best made of cast iron or steel highly polished, which allow the piston to move without, comparatively speaking, any friction. In a low-pressure cylinder of a modern marine engine, where the diameter is about one-and-a-half times the stroke, we find that the exposed surface of pistons

and covers is 50 per cent. greater than that of the working surfaces of the barrel, and it is to the former that I wish to direct your attention.

"Admitting that the greater the re-evaporation the greater will be the cooling of the surfaces, and, as a consequence, the greater the initial condensation, we must look more closely into the question whether the use of rough cast iron and steel or polished surfaces may have anything to do with the widely diverging results obtained from different engines of the same type, which apparently are well designed, and offer no peculiarities to account for something like 30 per cent. difference in economy.

"If we wish to shape a body which will readily absorb heat, we give it as large an area as possible to be exposed to the hot gases. A familiar instance is a 'Serve' tube. Also, if we wish to shape a body which will readily part with its heat to the substance which is to be heated, we do the same. For instance, the ribs and collars which are cast upon pipes for heating purposes.

"Passing to the rough surfaces of cast iron and steel, we may easily conceive that an almost imperceptible difference in the conformation of the surface may make an immense difference in the area exposed to the steam. Consider an element of piston area inclosed in an equilateral triangle; suppose equilateral triangles to be raised upon each of the sides, and joined at the top to form a tetrahedron; it is evident that the area now exposed will be three times that of the original triangle. If we divide the original triangle into four equal triangles, by drawing a line parallel to the base at the half of the height, and joining the points where this line intersects the two upright sides with the middle of the base, we can raise four triangular pyramids upon each of these parts, which will each expose an area of three times their bases, and collectively of three times the area of the original element. This operation can be indefinitely continued, with the same result, viz., that the exposed area is three times the area of the geometrical plane. A little consideration will show that the principle involved in this rudimentary case can be largely extended if we choose to build up other forms, and make surfaces analogous to, say, coke, or even the rough, rasping surface of some steel castings.

"With these considerations before me it struck me forcibly that in my own practice the most economical results were obtained with cast-iron pistons and covers, and the worst with those made of cast steel. Upon inquiry among my engineering friends, I found that those engines which were troubled with water in the cylinders had cast-steel pistons, and in cases where covers and pistons were of cast steel it had occurred that no diagrams could be obtained through the wetness of the steam."

The following experiment was made to test the re-evaporative power of polished and rough surfaces:

"A cylindrical cast-iron cup was bored out till the thickness of the metal was about $\frac{1}{8}$ inch; part of the outer surface was left as it came from the mould and another part opposite was polished. The cup was filled with water at temperatures from 170 deg. to 180 deg. Fahr., and it was found that the mean times required to re-evaporate the water from the steam jet condensed upon the surfaces were, respectively, 43 seconds for the rough and 83 seconds for the smooth face.

"I would now summarize the argument as follows: The greater the re-evaporative power of the material in contact with the steam, the greater will be the cooling of the surfaces, and the the greater the initial condensation; the re-evaporative power of a body is increased by an increase of surface; a fairly smooth surface may, nevertheless, have a very much greater area exposed than the geometrical plane, which it is taken to represent; the total condensation in a steam engine is a question of very thin films of water, and the structure of the surfaces exposed to the steam, merits our closest attention. As far as can be seen from the above experiments, a polished surface is far superior to that of ordinary cast iron as it comes from the mould, and a rough steel casting is probably about the worst material that could be chosen; it appears, therefore, advisable to carefully turn and polish as far as possible all parts of the steam cylinder which come into contact with steam performing work."

In accordance with this theory, the following experiment was carried out:

"I made a disconnecting paddle engine in which each wheel could be driven by a perfectly independent compound engine. In the starboard engine I had all internal parts turned and polished, and in the port one the pistons were left rough; unfortunately no time could be given for experiments, and I had to content myself with observing at sea that, whereas we were occasionally troubled with water in the port engine, the starboard one was apparently perfectly dry.

"In my experience excessive condensation has always occurred in combination with rough steel pistons and covers; in one case this was so marked that the top of piston would not give a card at all, which I attribute to the cover being of rough cast steel, and combining with the top of piston to absorb the heat of the steam.

"Torpedo boat engines, which have turned and polished pistons, and in some cases polished covers, are quite remarkable for the economy of steam at nearly all powers, and at all events are very free from water in the cylinders.

"These considerations lead to the conclusion that if we wish to make really high-class machinery, which will give the highest results in economy, we must turn and polish the surfaces of pistons and covers."

Messrs. Denny recently built a steamer, the *Queen Olga*, for the Russian Steam Navigation Company, which has engines designed to run as triple-expansion at full power, and as quadruple-expansion at reduced power. The cylinders are 21, 28¼, 47½ and 76 inches diameter by 54 inches stroke, and the first, or auxiliary cylinder, as it is called, is placed above the second cylinder, the two pistons working on the same rod. There are four boilers 15 feet 8 inches diameter, and 10 feet 7 inches long, working under a pressure of 170 pounds. On trial at full power, the engines developed 3,500 I.H.P. when making from 79 to 82 revolutions per minute, giving the ship a speed of 14½ knots. When running under half power, as a quadruple-expansion engine, the revolutions were 60, but no data are available as to the relative economy of the two methods of working.

The ship is 360 feet long, 45 feet beam, and when tried, with 2,400 tons dead weight on board, had a draught of 19 feet.

It will be remembered that the arrangement of the engines of *Gunboat No. 7* is the reverse of this, the design being for a quadruple-expansion engine working with 250 pounds pressure at full power, and for a triple-expansion engine working with 160 pounds pressure at low power, a special coupling being provided for throwing out the first cylinder when working triple-expansion.

THE ATLANTIC RECORD.

The Cunard steamship *Campania* broke all previous records on her last westward voyage, having made the run from Daunt's Rock to Sandy Hook lightship in 5 days 9 hours and 29 minutes. The daily runs were 516 (from 12.35 P. M.), 528, 543, 525, 545 and 126, a total of 2,783 nautical miles, or at the rate of 21.49 knots per hour. The *Lucania* has made a passage from New York to Queenstown at the rate of 21.9 knots per hour, but she took a more southerly course, having steamed 2,911 miles and occupied 5 days 12 hours and 59 minutes in making the run.

SHIPS.

UNITED STATES.

Minneapolis.—The official full-speed trial of this triple-screw protected cruiser took place on the 14th of July, and was run over the same course that was used for the trial of the *Columbia*, a distance of 43.968 knots in each direction between Cape Ann and Cape Porpoise. Buoys were fixed to mark the course, and vessels stationed at intervals to take tidal observations. The actual time on the course was 3 hours 49 minutes and 2.28 seconds, giving a mean speed, corrected for tide, of 23.073 knots. The lowest speed over any of the divisions of the course was 22.01 knots on the run north, and 22.48 over the same division running south; the greatest was 24.01 running south and 23.80 over the same division going north. The least depth of water on the course is 19 fathoms.

This remarkable performance easily puts the *Minneapolis* at the head of the crusier class in point of speed, exceeding that of the *Columbia* by more than a quarter of a knot, and it is the more remarkable from the fact that the trial was commenced 16 minutes before the time agreed upon, before steam had been raised to the required pressure, and that the air pressure was only one inch of water.

Owing to unavoidable circumstances a detailed account of the trial cannot be given in this number of the JOURNAL, but will be reserved for the next one.

The *Minneapolis* is of the same dimensions as the *Columbia*, and differs from her in outward appearance in having two smoke-pipes instead of four. The machinery of the two is almost exactly alike, the principal difference being in the boilers. Of the latter, six are 15 feet 9 inches diameter and 20 feet long, and two 15 feet 3 inches diameter and 18 feet long, and contain 1,456 square feet of grate and 48,194 of heating surface. There are,

in addition, two auxiliary boilers with 64 square feet of grate and 1,952 of heating surface, but they were not used on the main engines during the official trial.

The screws are three-bladed, and of the following dimensions:

	Center.	*Starboard.*	*Port.*
Diameter, feet	14.0	15.0	15.0
Pitch, mean, on trial, feet	21 5	22.0	22.0
Helicoidal area, square feet	53 28	53.7	53.7
Immersion of top of propeller blades, feet	9.12	5 46	5.46

The following are the average results of the trial:

Mean draught, feet and inches	22–6
Displacement, tons	7,387
Speed, knots	23.073
I.H.P. main engines	20,366
main engines and auxiliaries	20,812
Steam pressure in boilers, pounds	150.32
Air pressure in inches of water	0.998

		Center.	*Starboard.*	*Port.*
Steam pressure in H.P. steam chest		148 1	150.7	149.
I.P. steam chest, absolute		72.5	74.8	72.9
L.P. steam chest, absolute		24.6	26.3	27.4
Vacuum, inches		25.31	25.11	24.76
Mean effective pressure in cylinders	H.P.	51.91	49.32	46.39
	I.P.	31.20	34.53	28.52
	L.P.	15.04	13.33	13.90
Cut off in fraction of stroke	H.P.	.62	.66	.67
	I.P.	.71	.71	.72
	L.P.	.70	.69	.70
Revolutions per minute		132.19	131.95	133.1
Piston speed in feet per minute		925.33	923.65	931.7
Double strokes of air pumps		13.7	15.0	18.6
Revolutions of circulating pumps		159.4	240.5	168.8
I.H.P., main engines		6,586.72	6,560.69	7,218.82
air pump engines		8.39	10.91	13.86
circulating-pump engines		18.45	18.00	17.56
feed pumps			40.94	
blower engines			263.04	
other auxiliaries			54.92	
total for all machinery			20,812.	
per square foot of G.S.			14.29	

During the trip back from Boston to the Capes of the Delaware, three runs, each of eight hours duration, were made at

low speed, one with the center engine in operation and the side screws disconnected, one with the starboard and port engines and the center screw disconnected, and the third with the three engines in operation. The original intention was to make each of these trials at a speed of about 12 knots, but owing to foggy weather, it was necessary to reduce the speed frequently during the runs with one and with two screws, and at times during the run with three screws; so that the data obtained are of little value in determining the relative efficiency of the three methods of propulsion at low speed.

The general results were:

	One.	*Two.*	*Three.*
Number of screws in use			
Speed by patent log, knots	9.73	10 01	11.6
Revolutions, center	74.1	40.*	65.4
starboard	44.*	64.4	65.7
port	38.8*	62.7	65.9
I.H.P.	1,928	2,016	2,532
Coal per hour, pounds	4,629	4,773	6,481
Mean draught, approximate, feet	22.2	22.1	22.0

* Uncoupled.

Other trials were made with each of the side screws in operation separately, the results being:

	Starboard.	*Port.*
Revolutions	90	90
Speed	13.23	12.23
Helm angle, degrees	6.5	6.5
Revolutions of uncoupled screws	Port 50 St'bd 49; Center 44 Center 46	

The trials were of short duration, and the speed was measured by patent log.

Texas.—Dock Trial.—A dock trial of the machinery of this vessel was made on the 12th and the 15th of May at the Norfolk Navy Yard, to determine whether it is well built, and in conformity with the plans and specifications. It was made with each engine separately, the first trial with the starboard engine and the two starboard boilers, and the second with the port engine and the two port boilers in use. Each trial was of four hours duration, and was made under moderate forced draft, burning Pocahontas semi-bituminous coal. The conditions of the trial

were that the mean effective pressure for each trial should be equivalent to a mean pressure of 37 pounds per square inch referred to the low-pressure cylinder. But for the poor vacuum obtained, the trials may be considered highly satisfactory. The air pumps are independent, vertical bucket pumps, operated by a compound engine, and it was necessary to run them at a very high speed in order to get the vacuum they did and not run the risk of having them stop. Examination of the machinery subsequently to the trials showed it to be in very good condition.

The mean results for each trial are given below:

	Starboard.	*Port.*
Date of trial	May 12, 1894.	May 15, 1894.
Duration, hours	4	4
Steam pressure in boilers, per gauge, pounds	152.5	149.9
at engines, per gauge, pounds	148.1	145.0
at 1st receiver, absolute, pounds	82.9	74.0
at 2d receiver, absolute, pounds	27.7	32.5
Vacuum in condenser, inches	21.2	21.25
Revolutions of main engines, per minute	85.8	88.1
Mean effective pressure in cylinders: H.P.	46.98	48.96
Mean effective pressure in cylinders: I.P.	33.26	26.66
Mean effective pressure in cylinders: L.P.	18.25	19.35
Mean effective pressure in cylinders: Equivalent on L.P. piston	42.25	40.93
I.H.P.: H.P.	794.69	847.95
I.H.P.: I.P.	1,137.06	926.89
I.H.P.: L.P.	1,468.40	1,595.59
I.H.P.: Aggregate	3,400.15	3,370.43
I.H.P.: Air and circulating pumps	60.53	58.19
Cut-off in fraction of stroke, H.P.	.65	.66
Cut-off in fraction of stroke, I.P.	.65	.67
Cut-off in fraction of stroke, L.P.	.65	.66
Temperature in engine room	85.2	90.5
of injection water	71.0	71.0
of discharge water	113.8	107.0
of feed water	125.0	135.0
Double strokes of air-pump engine	198.9	179.0
Revolutions of circulating-pump engine	220.4	202.
Revolutions of blower engines	418.3	368.
Air pressure in inches of water	1.2	0.95
Grate surface in use, square feet	266.5	266.5
Heating surface in use, square feet	8,456.0	8,456.0
I.H.P. per square foot of grate	12.98	12.86

Montgomery.—During the trial of this vessel on the third of July, just as the speed of the engines was being increased preparatory to a run at full speed, a heavy thumping noise was heard in the port high pressure crank pin, and before steam could be shut off that engine the high pressure cylinder and cover were broken, and other damage of less consequence done to the connecting rod and crank pin brasses. The accident was occasioned by the slacking back of the set screw of one of the connecting rod nuts at the crank pin end, which allowed the nut to work off the bolt, thus throwing the strain on the opposite bolt, breaking it square off and bending and breaking the loose one; the connecting rod being thus released, the steam pressure forced the piston through the cylinder cover, demolishing it, and breakthe flange off the cylinder.

While waiting for a new cylinder, the port engine has been converted into a compound one, and the vessel continued in commission.

Oregon.—This battleship, building by the Union Iron Works, had a preliminary trial of her machinery in San Francisco Bay on the 16th of August; it was of four hours duration, everything working well, the revolutions of the engines not exceeding 96. Her official trial will probably not take place for several months, as her armor has not yet been put in place.

Massachusetts.—A sister ship to the *Oregon*, building by the Cramp Company, has had satisfactory dock trials of her machinery, and is now awaiting the completion of her armor.

CHILI.

Blanco Encalada.—A twin-screw protected cruiser, built by Armstrong, Mitchell and Co. She is a steel vessel, sheathed with wood and coppered, and is of the following dimensions:

Length on water line, feet,	370.
Breadth, feet,	46.5
Draught, mean, feet,	18.5
Displacement, tons,	4,500.

She has a protective deck which extends the entire length of the ship, varying in thickness from 1¼ to 4 inches, and has a

double bottom. Her engines, constructed by Humphrys and Tennant, are of the triple-expansion type, each with four cylinders.

She had two trials, one of 12 hours' duration under natural draft, during which she made six runs over a measured mile, the mean speed for which was 21.75 knots, with about 11,000 I.H.P. On the forced-draft trial the mean speed on the measured mile was 22.78 knots with 14,500 I.H.P.

Her battery is a very heavy one for a vessel of her size and speed, comprising two 8-inch, ten 6-inch, twelve 3-pounders, ten 1-pounders and two Gatlings. She also has five torpedo tubes.

Of her gunnery trials, "Engineering" says:

It should be remarked that the guns composing her main armament are of a peculiarly powerful type. The 8-inch guns are of 40 calibers length, and have a velocity of 2,260 f. s. with a 210 lb. projectile, and a very moderate chamber pressure. So improved is all the breech mechanism and powder supply arrangements, that the guns may be called quick-firers. This nomenclature was certainly borne out during the trials, when, with a crew that had never fired a round from the gun on any former occasion, four rounds were fired (the powder being supplied from the magazines) in 62 seconds; in fact, the man who has to aim the gun need never be kept waiting for the loading. The 6-inch quick-firing guns are mounted on pedestal mountings; they are also of 40 calibers length, and have a velocity of 2,500 f. s., with a 100-lb. shot. During the trials twenty rounds were fired from the 6-inch guns, and eight rounds from the 8-inch. Four rounds were also fired from each of the small quick-firing guns. No defect of any kind was experienced.

It should also be mentioned that to thoroughly try the structure of the ship, the after 8-inch gun was fired in line of keel horizontally, and the foremost 8-inch gun was fired ahead with elevation varying from the horizontal 4 degrees for five rounds, four of which were against time. This severe ordeal did not even crack the paint in the decks below the muzzles of the guns, and the only damage reported during the whole of the trials was

one broken rail, which had accidentally been left in a position of danger, and a little glass from the chart house.

Before the gunnery trials were undertaken, a torpedo was fired from each of the two port broadside tubes. These tubes are of new Elswick design, and are arranged for the use of cordite impulse, which has given such satisfactory results. A number of experiments has shown that the velocity of impulse of the torpedo is exceedingly regular, and that with a pressure of about 35 pounds in the tube a velocity of no less than 53 f. s. is obtained. The broadside tubes are capable of training through a large arc. During the trials the weather was beautifully clear, and much interest was taken in the long range of the guns when fired at high elevations. The 8-inch guns were fired at 15 degrees elevation, when the shot struck the water at a distance of 10,000 yards, but so clear was the weather that the spray thrown up by the shot was clearly seen each time.

ENGLAND.

Eclipse.—One of the latest type of second-class sheathed cruisers was launched on the 19th of July. She is of nearly the same dimensions as the *Olympia*—

Length, feet,	350.
Breadth, feet,	53.5
Draught, mean, feet,	20.5
Displacement, tons,	5,600.
Coal carried on above displacement, tons,	1,000.
I.H.P., natural draft,	8,000.
forced draft,	9,600.
Speed, knots,	19.5

Her protective deck is 2½ inches thick, and the conning tower 6 inches, besides which the top of the cylinders are protected by a 5-inch shield, as is usual in British vessels of this type. Her principal armament comprises five 6-inch and six 4.7-inch rapid fire guns, and the secondary battery twelve machine guns, besides which she has three 18-inch torpedo tubes. Her complement is 436 officers and men.

Flora.—A second-class sheathed cruiser of the *Astræa* class, described on page 518 of Volume V, recently completed her trials, making 19.3 knots under natural draft, and 20.43 under forced draft, the speed in each case being measured by patent log.

She is 320 feet long, 49½ feet beam, 19 feet mean draught, and, with 400 tons of coal on board, has a displacement of 4,360 tons. Her principal armament comprises two 6-inch and eight 4.7-inch guns. Her machinery was built by the Naval Construction and Armaments Co., Barrow-in-Furness.

Charybdis, another vessel of the *Astræa* class, has had satisfactory trials, the natural-draft one giving the following results:

Steam pressure in boilers,	141.7
Air pressure,	0.57
Vacuum,	27.2
Revolutions per minute,	127.7
I.H.P.,	7,109.
Speed, by patent log,	19.3
Mean draught of water, feet,	19.

The forced-draft trial gave 9,136 I.H.P., and a speed of 20.5 knots by patent log.

Daring.—This torpedo-boat destroyer was described on page 204 of the present volume, and illustrations and description of her boilers on pages 156 to 161. Since her launch, on the 24th of last November, the contractors, Messrs. John I. Thornycroft and Co., have made a number of experiments on her with different screws, and the result of her final trial was a higher speed than has yet been reached by any vessel afloat.

She is 185 feet long, 19 feet beam, 13 feet deep, and on her final official trial, drawing 5 feet 7½ inches forward and 6 feet 11¼ inches aft, displaced 240 tons. Her armament consists of one 12-pounder rapid-fire forward, three 6-pounders and three 18-inch torpedo tubes, one built in the bow and two swiveling in broadside.

The following description of the boat is chiefly from "Engineering":

The vessel shows a fine bold side for her inches, having a freeboard of 7 feet, and has, of course, a flush deck. The height above water makes her a dry vessel when running at speed, as was proved during the trial. There was, indeed, nothing that could be described as a heavy sea on during the trial, but the waves were quite big enough to have sent a good deal of loose water flying across the deck, when steaming with the wind ahead, had the boat not been well designed in this respect. The flush deck extends to the conning tower forward, beyond which there is the usual turtle back over the forepeak, as in the ordinary first-class torpedo boat design. The arrangement of the *Daring*, however, differs from the torpedo-boat design in regard to a breastwork at the end of the turtle back and extending right across the deck. This affords an excellent shelter to those forward when steaming head to sea.

The *Daring*, like all her sisters, is a twin-screw boat, and there is one important fact in regard to the business of boat destroying, for which the vessels are intended, should they ever be called upon for practical work: The draught of the larger single-screw torpedo boat is 7 feet 6 inches, whilst the *Daring* draws but 7 feet, so that in action the smaller craft would not be able to take advantage of shallow water as a haven of safety. The vessel has what should be described as the Thornycroft stern, although it is sometimes flippantly alluded to by the irreverent as the 'flat-tailed' or 'inverted pudding basin' type. The breadth is carried well aft on the water line, though there is a good deal of tumble-home at this part, and the end being round, the view end-on is somewhat unusual. In order to secure an easy delivery the floor is sloped up from near amidships to the extreme end, where it is only just immersed, so that the model may be described as that of a horizontal wedge aft and a vertical wedge forward. This principle of design, although somewhat unusual in steamers, is well known to give very good results, both in regard to speed and rough-weather qualities, for small sailing vessels, as may be instanced by the northeast cobles and the New England fishing schooners. In the Thornycroft design, however, the floor aft is not only flat, but actually concave in the 'twartship direction. This,

in conjunction with the tumble-home, gives a sharp knuckle or chine where the side joins the bottom. It has been objected that in a heavy sea the falling of the stern would cause a heavy blow on the bottom, but in practice this has not been found to be the case, as the boat in her passage appears to cut through the waves and smooth the water sufficiently to overcome this difficulty. The two propellers are carried under the arched bottom, the shafts projecting through, and there are two suspended rudders, one hung on each quarter in the same 'thwartship line as the propellers, to which they thus act as guards.

The engines of the *Daring* and her sister vessel the *Decoy*, now also finished, are an interesting feature, being of a new type designed by Messrs. Thornycroft for these vessels, and the first-class torpedo boats just completed. They are of the three-stage compound type, and have cylinders 19 inches in diameter for the high-pressure engine, 27 for the intermediate, and a pair of 27 for the low-pressure. The stroke is 16 inches. Each set of engines may be said to be divided into two parts. Thus the high-pressure and intermediate cylinders form one-half, whilst the two low-pressure cylinders form another. The cylinders are inclined from the vertical alternately to the right and left. Piston valves are not used, but the larger surfaces of the flat slide valves are balanced to relieve the pressure on the back. Following the line of the engines, we have first the eccentrics and link motion which work the slide valve of the high-pressure cylinder, which are run on an overhung part of the shaft. There is next a main bearing, after which come the high-pressure and intermediate cranks. Immediately aft again is another main bearing, and then the eccentrics for the intermediate slide valve, which are followed by the forward low-pressure eccentrics. Another main bearing follows, and then the two low-pressure cranks, after which another main bearing, and then the after low-pressure eccentrics, beyond which is the thrust block. The inclined position of the cylinders enables the axes of the cylinders to be brought close together in a fore-and-aft direction, and advantage is taken of this to introduce an arrangement of cranks that is certainly unusual for torpedo boat practice. In place of

there being a seperate two-web crank for each cylinder with a length of shaft between, there are three webs to the two crankpins, the center web being common to both, and receiving a crankpin at each end. The center web has naturally to be twice as long, or nearly twice as long, as each of the others—the cranks being almost opposite to each other—and it may be said, therefore, that the cranks are double-webbed. It will be seen that by this arrangement there is a considerable gain in fore-and-aft space occupied, no inconsiderable advantage in craft of the torpedo boat type, where machinery occupies all the best part of the boat.

Another notable feature in these engines is the manner in which the standards are arranged. The majority of us remember the revolution Mr. Thornycroft wrought in the design of engine framing by the introduction of wrought-iron or steel pillars and diagonal bracing, a principle which was first applied, if we remember rightly, in the case of the *Miranda*. The principle adopted in the little river yacht found universal application in fast-speed vessels, and has influenced all classes of marine engines. The design may now be said to have become classic, as it is superseded in the *Daring's* engines by a still more open description of framing. In the new arrangement the main standards are placed close to the fore-and-aft center line of the cylinders, and may be described as prolongations of the main bearing bolts; indeed, they serve the latter purpose, as the keeps of the main bearings are held in position by means of nuts screwed on to the bottom end of the standards or bearing bolts. By slacking up the nuts the keeps can be raised, and the bearings are thus exposed. The standards are stiffened by inclined struts. There is no bedplate to these engines in the ordinary acceptance of the term, but round one side, extending from end to end, there is a continuous steel casting connecting the main bearings. On the other side there are, to each set of engines, two flat steel castings which run from one pair of standards to another, and are attached to the bottom ends of the crosshead guides. In this way there is one of these deep steel ribs to each half of the set of engines, namely, one embracing the high and intermediate

cylinders, and one common to the two low-pressure cylinders. There are two air pumps, worked by side levers, to each set of engines, all other pumps being entirely independent. The question of vibration has been kept in view in these engines, the design being worked out so as to reduce the unbalanced forces to the smallest degree possible. Judging by results, Mr. Thornycroft has been entirely successful in this, for the steadiness of the boat when running at her highest power was almost as remarkable a feature as her extraordinary speed; indeed, the usual phenomena in this respect that formerly accompanied the running of lightly-built high-powered vessels were almost absent, there being apparent none of the usual periods of synchronization of hull and engines at varying revolutions, excepting, perhaps, to a small extent at a lower speed. It would seem, indeed, that the boat had nothing to synchronize with.

The boilers are three in number, and contain 189 square feet of grate and 8,892 of heating surface. They are placed in two stoke-holds, the forward one having two boilers and the after one a single boiler. The whole of the feeding is done by one Weir's pump placed in the engine room, there being, however, additional pumps to be used in case of accident. In order to solve the difficult problem of feeding a group of water-tube boilers from one pump, Messrs. Thornycroft have devised an automatic feed control. Throughout the trials there was no difficulty in regard to the distribution of feed, and it would seem that Mr. Thornycroft may, therefore, be congratulated on having solved a very knotty problem.

The screws are three-bladed, 6 feet 2 inches in diameter and 8 feet 9 inches pitch, each with a helicoidal area of 13.5 square feet. The blades are of phosphor bronze, keyed into the boss in a novel manner, it having been found by experience that phosphor bronze requires somewhat different treatment to steel.

The contractor's final preliminary trial took place on the 26th of June. A numerous party was on board, 73 all told, and with 19 tons of coal in the bunkers the displacement was 228.6 tons.

The vessel left Chiswick by the early morning tide, and, calling at the Temple Pier, took on board the guests who had been

invited to witness the trial. Easy steaming and moderate speed was maintained until the Lower Hope was reached, and down Sea Reach a fast spin was tried to see that all was working well. On reaching the Maplin mile the revolutions were reduced to a mean of 91 per minute, at which rate of steaming the observed speed was 7.86 knots. The engines were then opened out until they were running at 175 revolutions when the speed was 14.2 knots. At 238 revolutions the speed was 18.3 knots. After this the revolutions were raised to 322 per minute, with which a speed of 23.4 knots was attained. Three runs were made at each of the stated revolutions.

The final supreme test of the vessel was now to be made. The engines were fully opened out, and a short run was made in order to raise steam to full pressure and work the engines up to their best speed. The boat was once more put on the mile against the tide, which, however, did not appear to be running with very much force, to judge by the appearance of the Admiralty buoys which serve as steering marks. The steam pressure on this run was 200 pounds, the mean revolutions were 382.9 per minute, and the time occupied between the section posts was 2 minutes 7.5 seconds; the speed being, therefore, 28.214 knots. The boat was turned and the second run was entered upon, the steam being again 200 pounds. The time occupied in making this trip was 2 minutes 6 seconds, the revolutions averaging 385.7 per minute, the speed, therefore being 28.571 knots. The vessel was once more turned and the third run was entered upon with the safety valves from both boiler rooms blowing. The time occupied on this run was 2 minutes 3 seconds, the revolutions were 395.1 per minute, and the steam pressure 215 pounds. The speed, therefore, reached the hitherto unequaled rate of 29.268 knots. This run was made against tide, doubtless a slack one, but certainly there was some current meeting the boat. It will be seen that the mean of the three runs gives a speed of 28.656 knots, and 384.3 revolutions. The guaranteed speed of the boat was 27 knots.

It is to be regretted that this last trial, at full speed, was not extended to the usual six runs on the mile, for, had it been so,

there is little doubt that the mean speed would have been much increased. We think that too little time was taken in working up the rate of steaming before going on the full-speed trial, and this opinion is borne out by the records, and also by the fact that the safety valves were lifting on the last run. The afternoon, however, was drawing in, and Mr. Thornycroft had promised one of his guests that he should be back in time to fulfill an engagement. The indicator cards taken on the last run worked out to 4,842 indicated horse-power.

The official trial took place on the 19th of July, the boat being at her load displacement of 240 tons. This trial is thus recorded in the "London Times:"

"Reaching the trial ground off the Maplins, six consecutive runs were made on the measured mile, with the result that her previous speed record of June 26 was beaten, she having attained a *maximum* speed, on one of the runs made, of 29.364 knots, or 33¾ statute miles per hour. The runs were made with and against the tide, the *maximum* speed, as will be seen by the following tabulated statement, being made on the second run with the tide:

Runs.	Time.	Speed in Knots.	Mean of Runs.	
	m. *s.*			
1	2 4	29.032	28.111	Mean of speeds with tide = 29.065
2	2 12.4	27.191		
3	2 2.6	29.364	28.340	Mean of speeds against tide = 27.399
4	2 11.8	27.315		
5	2 5	28.8	28.246	Mean of means, 28.232 knots.
6	2 10	27.692		

"On an examination of this statement, it will be seen that the mean speed on each pair of runs, with and against the tide, which is the only correct way of summarizing them, was remarkably even, the results giving a true mean speed on the six consecutive runs of 28.232 knots, or 1.232 knots in excess of that contracted for. This noticeable equality in the resultant speed attained by the ship is accounted for by the great regularity in the revolutions of the propelling engines, and in the

steam pressure in the boilers, which, for the six runs made on the mile, is shown in the following statement, giving a mean of 388.2 and 388.4 revolutions per minute in starboard and port engines respectively, and a mean pressure of steam at engines of 198 pounds per square inch:

Runs.	Revolutions.		Steam Pressure.
	Starboard.	Port.	Pounds per square inch.
1	379.8	380.8	190
2	395.1	392.9	200
3	390.0	389.5	200
4	389.2	389.2	200
5	386.4	386.9	198
6	389.0	391.0	200

"The engines throughout the whole of the trial — which covered some considerable time of continuous running after the measured mile trials were completed—although driven at a high speed, worked with the greatest smoothness and regularity, and without the least sign of heating in any of the bearings; and the boilers worked admirably, the absence of any flaming at the funnels or noise from the safety valves being specially noticeable, the stoking of the boilers having evidently been so well managed that the steam pressure was maintained just at the working pressure required. For the size of the boilers, their steam generating powers are remarkable, a total of 4,735 indicated horse power having been given off by the engines on one of the mile runs, the pressure of the steam at the time being 208 pounds per square inch, maintained with 3.15 inches of air pressure in the stokeholds. Worked out to its conclusions, this means that an indicated horse power was developed in them by 1.66 square feet of heating surface, and .04 of a square foot of grate area. On the completion of the machinery trials, the turning and manœuvring capabilities of the vessel were successfully tested, after which she returned to the river and proceeded up to Chiswick, where she will be finally completed by her builders."

The true mean speed, deduced from the runs on the mile, is 28.213 knots.

Ferret.—Another torpedo-boat destroyer, built by Laird Brothers. She is similar in general appearance to the *Daring* and the other vessels of this type which have heretofore been described in the JOURNAL.

Length between perpendiculars, feet, . .	194.
Breadth, feet, ...	19.25
Draught, mean, feet, ...	5.
Displacement, tons, ...	220.
Coal carried on above displacement, tons, .	26.
Bunker capacity, tons, ...	70.

The engines are twin-screw triple-expansion, with cylinders 19, 29 and 43 inches diameter by 18 inches stroke, with the condensers (there are two) placed forward of the engines, thus giving a fairly good passage about the machinery. The boilers are of the Normand type.

Her official trial took place on the 10th of July, and consisted of six runs on the measured mile, followed by a continuous run of three hours duration, the results of which are given in the following tables:

SIX RUNS ON MEASURED MILE.

Miles.	Steam.	Vacuum.	Revolutions.	Indicated horse power.	Speed in knots.	Means.	Second Means.	
1	170	25 2	351.	4,127	26.866			
						27.018		
2	168	25.	356.9	4,375	27.170		27.178	
						27.338		
3	173	24.8	363.3	4,542	27.506		27.581	27 612
						27.825		
4	175	24.8	369.6	4,671	28.125		27.820	
						27.816		
5	172	24.8	364.1	4,656	27.506		27.871	
						27.926		
6	175	24.5	373.2	4,672	28.346			
Means	172.17	24.85	363.	4,507	...	...	...	27.612

THREE HOURS' TRIAL.

Half-hours.	Steam.	Vacuum.	Revolutions.	Indicated horse power.	Speed.
1st	173	25.	350.1	4,321	
2d	175	24.75	362.3	4,636	Mean speed of ship, 27.51 knots.
3d	170	24.75	365.1	4,810	
4th	171	24.5	362 4	4,508	
5th	161	24 75	361.	4,334	
6th	163	24.75	359 5	4.334	
Means	169	24.75	360.07	4,490.5	27.51

Lynx.—A duplicate of the *Ferret*, has had her preliminary trials, making 26.5 knots.

Dryad.—A first-class torpedo gunboat, was launched October 25, 1893. She is 250 feet long, 30 feet 6 inches beam, 9 feet mean draught and of 1,070 tons displacement, with 100 tons of coal in her bunkers. Her machinery is by Maudslay Sons and Field, and is of the same type and power as the other vessels of this class. Twin-screw, vertical triple-expansion engines, with cylinders 22, 34 and 51 inches diameter by 21 inches stroke, working with a boiler pressure of 155 pounds, each engine in a separate room; four wet-bottom locomotive boilers, placed two forward and two abaft the engines, containing 182 square feet of grate and 6,420 of heating surface. The condensing surface is 4,000 square feet. The screws are 8 feet 3 inches diameter and 9 feet 6 inches pitch. The boilers, with water and all fittings, weigh 110 tons.

The engines were designed for 3,500 I.H.P., with which a speed of 19 knots was expected.

As mentioned in the last number of the JOURNAL, she has had considerable trouble with leaky boiler tubes, but her machinery has finally been accepted from the builders. On her natural-draft trial she obtained 2,621 I.H.P. with .67 inch air pressure, and made 16.85 knots per hour.

Halcyon.—A torpedo gunboat of the same type as the *Dryad*, with machinery by Messrs. Hawthorne, Leslie and Co., developed 2,590 I.H.P. on her natural draft trial, and made a speed of

17 knots. On her first full power trial, some of the main bearing bolts of the starboard engine broke, and after larger ones were fitted she made a successful trial with the following results:

Steam pressure, pounds,	147.
Air pressure, inches of water,	2.0
Vacuum,	24.8
Revolutions,	247.7
I.H.P.,	3,546.
Speed by patent log,	17.7

The engines are of the same dimensions as those of the *Dryad*.

Algerine and *Phœnix*.—Two new gunboats to be built at Devonport dockyard. They will have a speed of about 14 knots and will be of the following dimensions:

Length, feet,	185.
Breadth, feet,	32.5
Draught, mean, feet,	11.25
Displacement, tons,	1,050.
Coal carried on above displacement, tons,	160.

Their armament will consist of six 4-inch 25-pounders, and four 3-pounders.

Torpedo Boats.—No. 92, a single-screw boat built by Thornycroft and Co., with engines of the same size and type as those of the *Daring*, and two boilers of the *Speedy* type, recently completed her trials, making 24.522 knots. The following are some of her particulars:

Length, feet,	142.5
Breadth, feet,	15.5
Draught, extreme, feet,	7.5
Displacement (with 24 tons load), tons,	129.
I.H.P.,	2,600.
Steam pressure, pounds,	220.

No. 91, a similar boat, is credited with 23.71 knots on the mile, and 23.74 for three hours, and a maximum I.H.P. of 2,629.

No. 93, a similar boat, but fitted with twin screws, made 23.846

knots on the mile and 23.59 for three hours, the revolutions of the engines being respectively, 472 and 467. Her load draught is 5 feet 4 inches. Each of these boats carries three 3-pounders and three 18-inch torpedo tubes.

No. 97, built by Laird Brothers, is similar to the preceding boats, 140 feet long, and of 115 tons displacement. On her first trial she failed to make the contract speed of 23 knots, but succeeded on the second one, the results on the mile being:

Runs.	Steam.	Vacuum.	Revolutions.	I.H.P.	Speed.
1	176	24 5	355.3	1,658	23.337
2	172	24.5	364.3	1,682	24.
3	176	25	362 3	1.704	23 529
4	174	25.	367.2	1,732	24.161
5	167	25.	360.7	1,650	23.226
6	163	25	356.1	1,628	23.529
Means.....	171	24.8	361.1	1 675	23.709

And 23.35 knots for three hours.

Crescent.—This first-class cruiser has just completed a run to Australia and back, having steamed a distance of 25,267 nautical miles in 82½ days, and burned 6,290 tons of coal, or an average speed of 12¾ knots an hour on 76 tons of coal a day. Her best day's run was 369 miles, made during the run from Malta to Plymouth, England, the entire distance having been made in 5 days 22½ hours.

She is a sheathed vessel of the *Edgar* class, and was described on page 746 of volume V. Length, 360 feet; beam, 60 feet 8 inches; mean draught, 24 feet 6 inches; displacement, 7,700 tons; I.H.P., 12,000.

FRANCE.

Jauréguiberry.—Some of the details of this battleship were given on page 1064 of Volume V of the JOURNAL, and the most important are here reproduced with additional data:

Length, feet,	356.
Breadth, feet,	72.59
Draught, mean, feet,	26.57
Displacement, tons,	11,818.
Indicated horse power,	13,000.
Speed, estimated, knots,	17.
Armor, thickness amidships, inches,	17.72
forward, inches,	7.87
aft, inches,	9.84
depth of belt amidships, feet,	6.56
lower edge below water line, feet,	4.92
thickness of turret, inches,	14.57
Battery, 30-centimetre guns,	2.
27-centimetre guns,	4.
14-centimetre guns,	8.
Torpedo tubes,	6.

The engines are twin-screw triple-expansion with cylinders 43.3, 64.2 and 97 inches diameter and a stroke of 44 inches, designed to run at 97 revolutions per minute at full power, with 170 pounds pressure of steam at the engines. The H.P. and I.P. cylinders have piston valves, and the L.P. each two slide valves. The condensing surface is 17,860 square feet for both condensers. There are two vertical single-acting air pumps for each condenser, each with cylinders 29.5 inches diameter and 16.73 inches stroke, intended to run at 164 revolutions per minute. The circulating pumps, which are 49.2 inches diameter, are intended to be run at the same speed.

The crank shafts are in two sections, each 17.33 inches in diameter with a hole 7.87 inches diameter through them, the intermediate shafts 16.93 inches with the same size hole as the crank shafts, and the propeller shafts 19.29 inches, with a hole 14.17 inches diameter; the total weight of shafting is 88 tons.

The screws are of manganese bronze, four-bladed, 19 feet diameter and 20 feet pitch, and both weigh 19 tons.

The boilers are twenty-four in number, of the Lagrafel-D'Allest type, arranged in six fire rooms, with two smoke-pipes, and contain 1,085 square feet of grate and 35,092 of heating surface.

There are 213 tubes in each boiler; they are 3.15 inches diameter and 8 feet long between tube sheets. The water space in the boilers amounts to 2,882 cubic feet and the steam space to 2,942.

The working pressure is 213 pounds, from which the steam is reduced to 170 pounds at the engines.

The weight of the boilers alone is 235 tons, of the smoke boxes 99, and of the water in the boilers about 80 tons. The feed pumps, floors, pipes, etc., bring the fire room weights up to 551.7 tons, and that of all the machinery to 1,120 tons.

The other most important weights are:

	Tons.
Hull,	3,317
Armor and bolts,	4,008
Guns and torpedoes,	995
Coal,	700

Carnot.—This battleship, which was laid down as the *Lazare-Carnot*, was launched from Toulon dockyard on the 12th of July. She is 396 feet long over all, 380.4 feet on water line, 70.5 feet beam, 26 feet mean draught and has a displacement of 11,791 tons. Her coal capacity is 800 tons. She has a complete belt of steel armor, varying in thickness from 45 to 27 centimeters (17.7 to 10.8 inches), and above this a belt of 4-inch armor.

Her battery consists of two 11.8-inch guns in turrets, one forward and the other aft, with a height of 26 and 19 feet respectively above the water line; two 10.6-inch guns in turrets on each side; eight 5.5-inch guns, four on each side; four 65-mm., eight 47-mm. and ten 37-mm. rapid-fire guns. She has also six torpedo tubes. The turret armor is 14.6 inches thick.

Her machinery consists of twin-screw, triple-expansion engines, designed for 13,500 I.H.P. at full power, when running at 95 revolutions per minute, and for 9,000 I.H.P. and 85 revolutions with natural draft. The speed in the two cases is expected to be 18 and 17 knots respectively.

Steam is furnished by twenty-four Lagrafel-D'Allest boilers. The weight of machinery is 1,178 tons.

Coetlogon.—This second class cruiser, which was built by the

Compagnie Générale Transatlantique at St. Nazaire, and which commenced her trials in 1890, completed them during the latter part of May. On her trials in 1891, she made 17 knots for twelve hours under natural draft, and 19.15 under forced draft. After the latter trial, the Commission appointed to examine her machinery declined to accept it on account of the numerous breakdowns she had; so, she was sent back to the builders, who put in new engines, and again submitted her for trial. During one of the preliminary runs early this year the tubes in her boilers gave so much trouble that new ones had to be put in. She commenced her final trials in March and finished them the latter part of May, the speed with natural draft being 17.83, and with forced draft 20.6 knots.

Torpedo Boats.—The *Argonaute* and *Tourmente*, built by the Chantiers de la Loire, completed their trials recently, the former making 25.1 knots and the latter 24.6. They are 141 feet long, 16.3 feet beam, draw 6.25 feet aft, and displace 117 tons with 15 tons of coal on board. They have twin-screw triple-expansion engines and DuTemple boilers. Their armament is two 3-pounders and two 14-inch torpedoes.

An explosion and a fire occurred on board the submarine boat *Gustave Zédé* a short time since, resulting in the practical destruction of her accumulators. The accident occurred while charging them, and is supposed to have been due to short circuiting, and consequent ignition of the valvoline with which the accumulators are filled. The damage is estimated at $40,000, and is the second one of the kind that has happened to this boat.

AUSTRIA.

Satellit.—In connection with the I.H.P. and weight of machinery of this vessel given on page 394 of the last number of the JOURNAL, Mr. C. H. Wingfield, an employé of the Thornycroft Company, writing to "Engineering," says that the I.H.P. averaged about 4,200, and that "the two horse-powers vouched for by Herr Ziese were obtained while the engines ran at their utmost speed for a short period only, and that this was not on an official trial at all, but during a preliminary run, on December

11, 1892. On the *Satellit's* preliminary trials I understand the revolutions were 280, 284 and 286 for 4,237, 4,792 and 4,901 indicated horse power respectively. If this be so, the indicated horse power on the official trial, which took place on December 30, 1892, when I believe the revolutions were 275.4, was probably from 4,000 to 4,300, or, say, about 4,200, which should be substituted for 4,901, in order to compare with the *Speedy's* so-called maximum performance."

The displacement of the *Satellit* given on page 423 should be 540 instead of 800, and the mean draught 9 feet.

ITALY.

Sardegna.—This battleship recently completed her trials in the Gulf of Genoa, making 17.6 knots on 85 revolutions, and 13,000 I.H.P. with natural draft, and nearly 19 knots for upwards of five hours with forced draft, the engines making from 90 to 95 revolutions. For three hours the speed was 19.8 knots. She was subsequently tried at reduced power, burning both coal and petroleum refuse in her furnaces, as is now frequently done in ships of the Italian Navy, the result being satisfactory, and the conclusion reached that, when firing in this manner, about double the quantity of steam can be got from the boiler that can be by burning coal alone with natural draft.

Caprera.—A torpedo gunboat, built by Orlando Brothers, Leghorn, was launched recently. She is 230 feet long, 28.6 feet beam, 10.17 feet mean draught, of 840 tons displacement, and has been designed for a speed of 20 knots, with 4,000 I.H.P. The armament comprises two 4.7-inch, four 57-mm. and two 37-mm. rapid-fire guns and five torpedo tubes.

Governolo.—A gunboat of 1,235 tons, designed for a speed of 12 knots with natural and 15 with forced draft. She is 186 feet long, 34.1 feet beam and 13.78 feet draught. Her armament is four 4.7-inch, four 57-mm. rapid-fire and four machine guns.

Umbria.—A protected cruiser of the *Dogali* class, made 18 knots under natural draft, with 5,000 I.H.P. She is 262.4 feet long, 39.4 feet beam, and of 2,280 tons displacement on a mean draught of 15 feet.

SPAIN.

Infanta Maria Teresa.—This belted cruiser, one of three built at Bilbao, was described on page 960 of Volume V, but as complete description with illustrations of the ship and machinery have since been given in "Engineering," the most important data are here reproduced, together with the additional information available:

Length over all, feet,	364.0
between perpendiculars, feet,	340.0
Breadth, extreme, feet,	65.2
at upper deck, feet,	62.25
Depth from under side of upper deck to top of keel, feet,	38.0
Draught, forward, feet,	20.5
aft, feet,	22.5
mean, feet,	21.5
Displacement at above draught, tons,	6,890.
Wetted surface, square feet,	27,000.
Metacentric height, completely equipped, feet,	4.96
Metacentric height, without stores, coal, or ammunition, feet,	5.49
Speed, natural draft, knots,	18.5
forced draft, knots,	20.24

The weight of the vessel is made up as follows:

Hull (excluding armor), tons,	3,639.
Armor (including the belt, 216 feet long, 5.5 feet deep, and 12 inches thick), tons,	943.
Machinery (including boilers and water), tons,	1,230.
Coal in bunkers (at 21.5 draught), tons,	420.
Armament, ammunition, etc., tons,	456.

Stores, etc., bring the total to the designed displacement, 6,890 tons. It may be added here, however, that the vessel has bunker capacity for 1,100 tons of fuel.

The hull is built with longitudinal and transverse framing. The center longitudinal web girder is 3 feet deep, and is built up of plates and angles, while on either side are four similar girders, but lessening in depth towards the bilge, where they are 2.5 feet deep. There are the usual intercostals, and plating on inner and outer sides. This double bottom construction, formed on the cellular principle, extends up the side of the ship to near the water line, where the fourth girder forms a shelf 3 feet wide to support the armor belt of 12 inches thickness, on the top of which again is the protective deck of 3 inches and 2 inches thickness. The armor belt extends for a distance of 216 feet amidships, and is 5.5 feet broad. The plates are 12 inches thick, and are secured by 3½-inch bolts, the backing of teak being 6 inches thick. For additional protection coal bunkers are arranged on either side of the ship, in the wake not only of the boilers, but also of the machinery. The capacity of these bunkers is 45,000 cubic feet, capable of carrying 1,100 tons of coal. There are eleven main transverse bulkheads, the boilers being fitted in two separate compartments, while the two sets of engines, driving the twin screws, are also separated by a bulkhead running longitudinally. In addition, the principle of water-tight flats has been adopted, and in this connection it is interesting to note that the bulkheads are fitted with a hinged water-tight door, the invention of Mr. James S. Clark, manager of the shipbuilding yard. The arrangement of the door is such that by the working of one lever fitted on the door, it can be locked or unlocked at all parts. The hinged door is a great advantage, especially as it does not interfere with freedom of action on the deck on which the door is situated. Of course, greatest dependence is placed, in action, on the protective deck, which extends from stem to stern, curving down aft and forward to considerably below the water line. This deck is 2 inches thick on the horizontal parts, and 3 inches on the sloping parts, and covers the under-water structure, which, as already pointed out, has two outer skins separated by 2.5 to 3 feet of space. Under this deck, of course, are all the vitals of the ship—the stores, the magazines, steering gear, and the machinery—but the tops of the cylinders project a short dis-

tance above the level of the deck. The deck, however, is raised locally and, instead of being of double thickness of steel plates, as in other parts, the protection is of 6-inch compound armor plates.

Dealing now with the navigation of the ship, a word may be said of the accommodation for the crew. The berths are entirely on the main deck, which runs in an unbroken line fore and aft. The commander's cabin is, of course, aft, with the usual stern walk, the officers being further forward, while the engineering officers are immediately aft the machinery, the firemen and crew being forward. The ship is navigated from the bridge, or from the conning tower below it. This latter is protected by compound armor 12 inches thick, and is fitted with engine and steering telegraphs, steam steering wheel, and voice tubes to all parts of the ship, and all other apparatus necessary for working and fighting the ship when in action. The telegraphs, etc., are duplicated on the navigating bridge, placed above the conning tower. The steering gear can be worked by hand and steam, the steam-steering engine being of sufficient power to put the helm from hard-a-port to hard-a-starboard in 30 seconds, when the vessel is proceeding at full speed. The rudder, which is of wrought iron, is of the ordinary rectangular form, and has an area of about 160 square feet. The rudder head extends only a few feet within the vessel, just high enough for the attachment of a steel crosshead giving connection to the steering tiller, so that the whole of the steering gear is entirely below the protective deck. The ship was subjected to severe turning trials at 10 and 16 knots speed, and they established the efficiency and ease with which she could be handled. The following are the results:

	Degrees.		Knots.				Yards.
Rudder,	20;	speed,	10;	diameter of	turning	circle =	1,540.
"	30;	"	10;	"	"	" =	940.
"	20;	"	16;	"	"	" =	1,040.
"	35;	"	16;	"	"	" =	840.

Forward there is a powerful capstan, and a large warping winch aft. The principal ground tackle consists of two 90-cwt.

and two 30-cwt. ordinary anchors, two 80-cwt. stockless, and two kedge anchors of 14 cwt. and 8 cwt. respectively. The pumping arrangements and ventilation of the ship have been well looked after, and throughout the whole ship a proper and complete system of voice-pipe arrangements has been installed. The necessary fire pumps, with their connections and hose, have been provided as usual. As to stores, it may be stated that under the boiler compartments there is capacity for storing 490 tons of fresh water, while at various points on the main deck are water tanks, etc. Navigating stores and chain lockers are, as usual, forward.

She carries twelve boats, one 60-foot vedette, two steam launches, four sailing cutters, one 30-foot gig, one 28-foot whale boat, one 25-foot canoe and two dinghies.

The vedette boat has a compound engine 10 by 20 by 10, and has a speed of 17 knots. The steam launches are each 30 feet long; one is 8.5 feet beam and 3.75 deep, and the other 7.25 feet beam and 3.5 feet deep. Their engines are compound, 4.75 by 9 by 5 inches stroke, and the boiler 2.83 feet diameter and 4 feet long, working at a pressure of 130 pounds. The boiler contains 3.6 square feet of grate and 88 of heating surface, and is arranged to work under forced draft, each launch having an 18 inch fan run by a 2 by $1\frac{1}{2}$-inch engine. The screw is 28 inches in diameter, and has an area of 1.68 square feet. The speed of the launches is 8.5 knots when the engines are running at 440 revolutions per minute. There is a boat-hoisting engine on the spar deck with steam cylinder 9 by 7 inches stroke.

Coming now to the ship as a fighting machine, it should be stated at the outset that the ram, which is of steel, is strengthened by longitudinal plate girders, while the protective deck, which slopes downwards at the fore end towards the ram, materially strengthens the bow for ramming purposes. The vessel has a flush upper deck without bulwarks, with a breastwork within which the 6 inch quick-firing guns are installed. Within the extremities of the belts are the two barbettes, one at each end, these being built of compound armor 10 inches thick. The protection to the ammunition tubes, which are of a conical form,

rises from the protective deck, and is made of steel armor plates 8 inches thick. The barbettes are of circular section, and stand about 6 feet above the upper deck, with a strongly protected roof plate 4 inches thick. Each barbette contains a 28-centimetre breech-loading Hontoria gun. The gear for training the guns and hoisting the ammunition is situated below the protective deck. The turntable upon which each 28-centimetre gun is mounted is constructed of steel plates and angles strongly built. It is supported upon cast-steel rollers traveling on a cast-steel roller path at the level of the deck. The carriage, mountings, hydraulic machinery, etc., for training the guns, also the ammunition hoists and tubes, are supplied by Messrs. Sir Joseph Whitworth and Co. Between the barbettes on the upper deck is a superstructure or breastwork, within which are all the openings to the deck below. At each end of the breastwork is an ammunition hoist for the 14-centimetre quick-firing guns, which is worked from below the protective deck, and will greatly facilitate the work of serving the guns. The ship has in all eight torpedo tubes, and the principal armament is as follows: Two 28-centimetre guns, one forward and one aft, mounted in barbette turrets; ten 14-centimetre guns, two 7-centimetre guns, eight 57-millimetre Nordenfeldts, two 11-millimetre Nordenfeldts, and eight Hotchkiss. The forgings for these guns were sent from England, but they were turned and finished in the Astilleros Gun Factory, all the employés of which are Spaniards.

The engines are of the triple-expansion vertical type, driving twin screws, and were designed to develop 13,500 I.H.P. under forced draft. The cylinders are 42, 62 and 92 inches diameter, by 46 inches stroke, and are provided with fluid compressed steel liners. The cylinder covers, pistons and steam-chest doors are all of cast steel. The high pressure cylinders are fitted with piston valves, and the intermediate and low pressure with valves of the ordinary flat-faced ported type. The piston rods, which are $7\frac{3}{4}$ inches in diameter, are all fitted with metallic packing. The crankshafts are of the ordinary three-throw type, being made of best steel, the external diameter being $16\frac{1}{8}$ inches. The thrust blocks and collars are of cast steel, the latter being of the

horseshoe shape and lined with Magnolia metal. For each engine there are four lengths of straight shafting. The diameter of the intermediate shafting is 15¼ inches, that of propeller shafting being 15¾.

The propellers are of manganese bronze, three bladed, the pitch variable from 19 feet 2 inches to 22 feet 3 inches, and set at 20 feet 6 inches on trial. They are 16 feet 5 inches in diameter, and each has a surface 72.72 square feet. The hubs are 52 inches in diameter.

The main steam pipes are 18 inches in diameter, and are all of copper, the sheets having been brazed and wrought up in the coppersmithy. The reversing gear is of the ordinary all-round type, both hand and steam gear being provided. The main surface condensers, which are 10.67 feet long, are made entirely of brass, having a total surface of 14,600 square feet. Each condenser carries over 5,000 brass tubes ⅝-inch in diameter. The circulating pumps are driven by a separate engine, while the air pumps are driven from the low-pressure engine crosshead. In each engine room there is a 100-gallon Kirkaldy's distiller, each with its circulating pump, as also an evaporator on Weir's system, 5 by 4.25 feet, with a 4-inch cylinder feed pump. The exhaust steam from the auxiliary machinery is carried into two auxiliary condensers 8 feet long by 2.83 feet in diameter, one in each engine room, each condenser being provided with its own air and circulating pumps worked by entirely independent engines. These, like the main condenser, are made entirely of brass, and each contains about 700 tubes, having a total cooling surface of 900 square feet. There are two main feed pumps—Weir's system—one in each engine room, which have steam cylinders 15 inches in diameter and pump 11½ inches in diameter by 27 inches stroke.

There are four double-ended and two single-ended boilers in two water-tight compartments forward of the engines, the single-ended boilers being at the forward end of the forward boiler compartment, all arranged for working under forced draft on the closed fire room system. The combustion chambers of all the furnaces in each boiler are in common. The following are the principal data of the boilers:

Steam pressure, pounds,	150.
Diameter of boilers, feet,	15.25
Length of double-ended boilers, feet,	16.25
single-ended boilers, feet,	10.50
Furnaces, number,	40.
diameter, inside, inches,	39.
length, double boiler, feet,	6.9
length, single boiler, feet,	7.4
thickness, inch,	.484
Tubes, plain, number in each double boiler,	792.
number in each single boiler,	396.
thickness, B.W.G.,	No. 8.
stay, number in each double boiler,	280.
number in each single boiler,	140.
thickness, inch,	.188
diameter, inches,	2.5
length between tube sheets, double boilers, feet,	6.25
length between tube sheets, single boiler, feet,	6.75
Shell sheets, inches,	1.28
Tube plates, inch,	.81
Combustion chamber plates, inch,	.50
Front heads, inch,	.875
Longitudinal braces, diameter, inches,	2.56
spacing, horizontally, inches,	17.5
spacing, vertically, inches,	16.
Depth of combustion chamber, common, in double-ended boilers, feet,	3.48
Depth of combustion chamber, single-ended boilers, feet,	2.92
Grate surface, square feet,	845.
Heating surface, tubes, square feet,	22,270.
plate, square feet,	3,650.
total, square feet,	25,920.

For supplying steam to the electric light and steering gear engines there are two auxiliary boilers on the protective deck forward 8.88 feet long by 7.75 feet in diameter, with a duplex feed pump on the Worthington system. These boilers have furnaces 29 inches in diameter, and grates 5 feet 3 inches long, and have together 50.7 square feet of grate and 916 of heating surface. The feed pump has steam cylinders $4\frac{3}{4}$ inches in diameter, and water plungers $2\frac{3}{4}$ inches in diameter by 5 inches stroke. The usual stokehold arrangements for running under forced draft have been fitted, there being nine fans. Each is driven by an independent single-acting engine with a cylinder 7 inches in diameter by 5 inches stroke. The diameter of fans is 66 inches. There are four ash-hoisting engines, having two $4\frac{1}{4}$-inch cylinders by $5\frac{1}{2}$ inches stroke, and efficient means have been provided for handling the ash buckets under forced draft. There are also two fans for ventilating the ship, one forward and one aft. The cylinders are $4\frac{1}{2}$ inches in diameter by $3\frac{1}{2}$ inches stroke, and the fans 39 inches in diameter. There are two funnels 9 feet in diameter, the height from dead-plate to top of funnel being 69 feet. Following out the usual plan, to provide greater safety, the boilers are placed in two separate compartments, the bunkers being run along each side in the usual way.

The dynamos and engines are placed below the protective deck at the after end of the ship. The engines are of the vertical type, and at 225 revolutions the dynamos give an output of 300 ampères and 80 volts each. There are three powerful search lights of 24,000 candle-power each, and over 400 incandescent lamps of 16 candle-power each, distributed throughout the ship. There are two cargo lanterns, with eight 50 candle-power lights in each; also a complete system of signaling on the Ardois system.

The trials were run on the standardized screw method, the ship having been run over a measured base of 1.412 knots four times under natural and forced draft, after which the trials were made at sea, the results of which are given in the table. Upon completion of the sea trial the ship was again run over the measured base, and the mean revolutions taken as the revolutions per knot. The guaranteed speed under natural draft was 18 knots, with a

penalty of $15,000 for each tenth of a knot below this speed. The fans were running throughout the trial, the mean air pressure being $\frac{3}{16}$ inch.

	Natural draft, September 18, 1893.	Forced draft, October, 14, 1893.	Coal consumption, October 18, 1893.
Revolutions per minute	107	118	55
I.H.P. starboard, H.P	1.592	2,069	259.38
I.P.	1.553	2,290	211.38
L.P.	1,487	2,460	120 14
port, H.P.	1,626	2,075	273.65
I.P.	1,606	2.515	190.00
L.P.	1,635	2,313	176.65
Total	9,499	13.722	1,231 2
Steam pressure, pounds	140	143	110
Vacuum, starboard, inches	27	27½	26½
port, inches	27	27	27
Receivers, starboard, H.P., pounds	117	134	33
I.P., pounds, absolute	51	67	18
L.P., pounds, absolute	21	28	10
port, H.P., pounds	112	140	34
I.P., pounds, absolute	53	65	17
L.P., pounds, absolute	22	30	10
Mean pressure, starboard, H.P., pounds	46.2	54.4	14.7
I.P., pounds	20.7	27.6	5.5
L.P., pounds	9.0	13.5	1.42
port, H.P., pounds	47.2	54.6	15.45
I.P., pounds	21.4	30.4	4.9
L.P., pounds	9.9	12.7	2.08
Cut-offs, H.P., per cent	62.5	73	55
I.P., per cent	70	75	60
L.P., per cent	70	75	67½
Speed, knots	18.5	20.24	10
Duration of trials, hours	6	4	12

Indicated horse power per square foot grate area:

Natural draft	11.24
Forced draft	16.24
Indicated horse power (forced draft), per ton of total weight of machinery	11.16

The forced-draft trials were run on October 14, 1893. Six runs were first made on the measured distance, and thereafter two hours steady running to sea, concluding with four runs again on the measured distance, the total duration of trial being, therefore, 4½ hours. During the whole time the machinery worked exceptionally well. The average speed of 118 revolutions was kept up steadily, the highest revolutions counted being 120. The steam never fell below 140 pounds, and maintained nearly throughout a uniform pressure of 145 pounds, any excess of which the contract forbade. The mean air pressure in the stokehold was only 1 inch.

The speed was 20.25 knots, a quarter of a knot more than the contract called for. There was no premium, but the conditions of the contract were that for every tenth of a knot below 20 there should be a penalty of $5,000.

The fuel-consumption trials were run on October 18 and 19, 1893, the weather being very fine, but with a slight swell on the sea. The vessel was, according to condition of contract, to be run for at least 12 hours at a speed of 10 knots. The coal was to be put in bags to facilitate the calculation of the consumption, which, at 10 knots speed, might "vary, without giving rise to the imposition of fines, between 635 and 680 grammes per hour and horse power indicated, developed by the principal engines and the auxiliary apparatus connected therewith." The vessel started at 3 o'clock in the afternoon, and six runs were made on the measured distance of 1.412 knots, the cruiser thereafter continuing out to sea until 6 o'clock on the following morning, October 19. The coal was carefully weighed in the stokehold by the Arsenal experts and put into bags, each of 165 kilogrammes, and men were stationed at each of the bunker doors to prevent more coal being taken out than had been weighed. One single-ended and two double-ended boilers were employed to supply the steam to the main engines and to the auxiliary engines connected therewith, and for the auxiliary engines, (*i. e.*, steering-gear engine, electric-light engine, auxiliary circulating pumps and bilge pumps), one of the single-ended boilers was used,

but without any connection to the main engines. The consumption of the coal was carefully recorded for twelve hours in the case of the main boilers, and for eight hours in the case of the auxiliary boiler. The result was that the coal consumption for the propelling machinery was found to be 1.43 pounds per indicated horse power per hour, the engines meanwhile running at a mean of 55 revolutions and developing 1,231 horse power.

One single-ended boiler supplied steam to the following auxiliary engines: Two auxiliary circulating pumps, two fire and bilge pumps, steering-gear engine, electric-light engine and auxiliary feed pump, the total horse power developed being 197. The total coal consumed per hour in the single-ended boiler was 239.5 kilogrammes, which gives 2.68 pounds per indicated horse power per hour. The auxiliary engines above referred to are single cylinder engines, with the exception of the electric-light engine, which is compound, having cylinders 7½ inches and 13½ inches in diameter by 8 inches stroke.

The contract further stipulated for a trial with one screw, the rudder being thrown over to maintain a straight course, and the guaranteed speed being 12 knots. This trial, under natural draft, was run on October 19. The port engine was used, the rudder being from 12 to 15 degrees over. The mean speed of three runs on the measured mile was 12.56 knots, with the engine running at 85 revolutions. Steam was supplied from one single-ended and two double-ended boilers.

On the same day the starting, stopping and reversing trials were gone through, with the following results: From full speed ahead to stop, 7 seconds; from full speed astern to stop, 6 seconds; from full speed astern to full speed ahead, 7 seconds.

The contract also stipulated for a trial of the engines to ascertain the least number of revolutions required to enable the engines to work with perfect regularity with the valves sufficiently closed, with the normal vacuum in the condenser and with 145 pounds boiler pressure. This proved to be 13 revolutions, equal to a speed of ship of about 2½ knots. At this speed the engines were kept running for about 10 minutes.

JAPAN.

Fuji Yama.—The Japanese government has entered into contract with the Thames Iron Works, and with Armstrong, Mitchell and Company for the construction of two battleships similar in general characteristics to the *Royal Sovereign* class in the English Navy. The *Fuji Yama*, building by the Thames Iron Works, will be of the following dimensions:

Length between perpendiculars, feet, . .	370.
Breadth, extreme, feet, ...	73.
Draught, mean, feet, ...	26.5
Displacement, tons, ...	12,250.
Coal carried on above displacement, tons, .	700.
Bunker capacity, tons, ...	1,100.

She will have an armor belt about 225 feet long, 18 inches thick along the machinery space, and 16 inches beyond. The protective deck will be 2½ inches thick. The battery comprises four 12-inch and ten 6-inch guns, beside twenty-four of small caliber and six torpedo tubes. The 12-inch guns are mounted in pairs in barbettes protected by armor 14 inches thick. The engines will be of the triple-expansion type, and of 14,000 I.H.P., steam being supplied by cylindrical boilers. The speed expected is 18 knots.

MERCHANT STEAMERS.

City of Lowell.—This steamer, built by the Bath Iron Works, for the Norwich and New York Transportation Company, was described on page 1088 of Volume V. She has taken her place on the route, and has proved herself a worthy competitor of the fastest steamers on the Sound. On her builders' trial, she was twice run over a measured mile, and made it in 2 minutes 56½ seconds in one direction, and in 2 minutes 55 seconds in the opposite direction, giving her a mean speed of nearly 20½ miles an hour. The steam pressure was 150 pounds, and the revolutions 126 and 128 for the starboard and port engines respectively.

Petersburg.—A twin-screw steamer built by Hawthorne, Leslie and Company for the Russian Volunteer Fleet. She is divided into ten water-tight compartments, and has a double bottom extending her entire length. She is 460 feet long, 54 feet beam, 35 feet deep, and measures 6,000 tons. Her engines are each in a separate compartment, and have cylinders 34, 54 and 85 inches diameter by 51 inches stroke; the valves are operated by Marshall valve gear. Steam is supplied by seven double-ended boilers, with a total of 42 furnaces. As she has been built with a view to being used in time of war, she has accommodation for 1,500 troops, and her bunkers are arranged along the sides so as to give protection to the machinery. Her contract speed of 19 knots was reached on trial.

Moor.—This steamer, belonging to the Union Steamship Company, and engaged in the trade between England and the Cape of Good Hope, was recently lengthed 55 feet by D. and W. Henderson, Glasgow, making her 420 feet long, 45.5 feet beam and 32.5 feet deep. A portion of the additional space thus gained has been given up to two boilers 15 feet diameter and 10.44 feet long, and to coal, and the remainder, representing about 550 tons,

to cargo, the total increase being about 1,000 tons in displacement, and an increase of 6 inches in the draught of water.

The engines, except bed plates and some of the shafting were also changed, the new engines being 35, 56 and 90 inches diameter by 60 inches stroke. The diameter of shafting is the same as with the former engine, 17¼ inches, but the thrust surface has been increased from 1,334 to 1,953 square inches. There are now six double-ended boilers 13.64 feet diameter and 18 feet long, and two single-ended 15 feet diameter and 10.44 feet long, the furnaces in the former being 37, and in the latter 44 inches diameter, the addition of the two boilers resulting in an increase in grate surface from 360 to 490 square feet, and in heating surface from 10,557 to 14,901 square feet.

There are now two smoke pipes instead of one, and they are 90 feet in height above grates.

A new screw 19 feet diameter and 20.5 feet pitch was substituted for the former one 17.5 feet diameter and 25.5 feet pitch, so that the engine now runs at 85 instead of 70 revolutions.

A new condenser was put in, increasing the condensing surface from 5,490 to 7,500 square feet, and centrifugal circulating pumps substituted for the reciprocating ones.

On a measured mile trial the speed was upward of half a knot more than with the former machinery before the ship was lengthened, although the displacement has been considerably increased. This is the third set of engines that has been put in this steamer, her first ones having been compound, and the result of similar trials with the three sets of machinery were:

Engine	Compound.	First, Triple-Ex.	Second, Triple-Ex.
Displacement, tons	4.700.	5,090.	6,466.
Revolutions	69.5	70.	85.5
I.H P	4 340.	4.532.	6,071.
Speed, knots	15.25	15.93	16.52

Danube.—A steamer built by James and George Thomson for the Royal Mail Line, for service between Southampton, England, and the River Platte, has recently completed her trials. Her length over all is 435 feet, length between perpendiculars 420 feet, beam 52 feet, molded depth 35.4 feet, gross register tonnage

6,050, and at a mean draught of 21 feet has a displacement of 8,000 tons.

Her engines are of the usual inverted direct-acting triple-expansion type, with cylinders 38, 60 and 94 inches diameter and 66 inches stroke of pistons, designed for 8,000 indicated horse power on about 900 feet piston speed. The engine framing and bed plates are of cast iron, the tunnel shaft of wrought iron, and the other shafting of Vickers' steel, 19 inches in diameter. The air and circulating pumps are worked from the high and low-pressure crossheads. She has four double-ended boilers, 15 feet in diameter and 19 feet long, working under a pressure of 160 pounds; they each contain eight furnaces 39 inches in diameter and 6 feet 6 inches long, the total grate surface being 650, and the heating surface 17,300 square feet. There is, besides, a donkey boiler for supplying the auxiliaries.

On her official trial on the measured mile, she made an average speed of 17.42 knots, the detailed results of each run being:

	Pressure.	*Revolutions.*	*I.H.P.*	*Speed.*
1st, . . .	145	82.3	7,532	18.75
2d, . . .	150	82.1	7,488	16.00
3d, . . .	150	83.8	7,780	18.48
4th, . . .	155	83.9	7,971	16.67
5th, . . .	155	83.9	7,941	18.18
6th, . . .	150	83.3	7,828	16.98

She is a sister of the *Nile*, which was described in Volume V.

Seine and *Thames*.—Two light draught, twin-screw steamers, belonging to the London and Brighton railway and the Compagnie des Chemins de Fer de l'Ouest, engaged in service between Dieppe, France, and Newhaven, England. They were built at Havre by the Forges et Chantiers de la Meditarranée, under guarantee that the average speed for the run should not be less than 21 knots, including the time lost in slowing down and in getting headway on entering and leaving port. The *Seine*, which was the first one built, was allowed 3½ hours for the trip, but made it in 3 hours and 9 minutes; the *Thames* was allowed 3 hours and 20 minutes and made it in 3 hours and 3 minutes. The hulls of both are alike, and of the following dimensions:

Length over all, feet,	269
Beam, feet,	29.53
Depth, feet,	15.1
Draught, amidships, feet.	8.33
aft, feet	9.19
Displacement, tons,	1,020

The engines are of the vertical triple-expansion type and are alike in each vessel. Owing to the limited draught of water, the screw propellers, which are 9 feet in diameter are partly out of water, and are said to work satisfactorily even at the highest speed. The *Seine* has six cylindrical return tubular boilers, working at a pressure of 155 pounds; they contain 336 square feet of grate and 9,720 of heating surface, and furnish steam for 4,000 I.H.P. On the *Thames* there are twelve Belleville boilers working at a pressure of 242 pounds, containing 374 square feet of grate and 11,000 of heating surface, and the I.H.P. is about 4,500.

Both vessels are, of course, built of steel, subdivided into water-tight compartments, and are fitted with ample accommodation for 700 passengers.

A third vessel building by Denny Brothers for the same line was launched in April, but broke down on her trial about two months ago.

Duke of York.—A twin-screw steamer built by Denny Brothers, of Dumbarton, for the London and Northwestern Railway Company. She is 310 feet long on the water line, 37 feet beam, and 17 feet deep, and is intended chiefly for passenger traffic, there being sleeping accommodations for 230 first-class passengers.

The engines are of the triple-expansion type, with cylinders 23, 35½ and 53 inches diameter by 33 inches stroke, and steam is furnished by two double-ended boilers working under forced draft on the closed fire room system. On her acceptance trial, with ½ inch air pressure in the fire rooms, the speed was 19.06 knots.

La Marguerite.—A paddle-wheel steamer built by the Fair-

field Shipbuilding and Engineering Company, to run between London and Boulogne. Her dimensions are:

Length between perpendiculars, feet, . .	330
Beam, moulded, feet, ...	40
over guards, feet, ...	73
Depth, moulded, feet, ...	21.5
Gross tonnage, ...	2,205
Draught, mean, feet, ...	8.90
Displacement, tons, ...	1,868

The hull is of steel, and is divided into eleven water-tight compartments, special attention having been given to strength and rigidity on account of the character of the service she has to perform. To facilitate the manœuvring in port a bow rudder has been fitted, and this takes the form of the run of the boat. It is operated by screw-steering gear fitted on the upper deck and worked with the usual wheel. A simple locking arrangement is fitted for keeping it in normal position when the vessel is driving ahead. The after rudder is of large area, and is fitted with steam steering gear, placed in the engine room and operating the tiller head by chain gear. Steam capstan forward and aft and windlass forward enable the ship to be easily warped up against the piers, and the telegraph arrangements from the captain's bridge are very complete, so that all the navigating appliances are of the modern type.

The vessel has four decks, the promenade, upper, main and lower decks. The rooms on the lower deck occupy the full width of the ship, and large square windows pierce the shell plating. To conserve the strength, these square spaces are divided into four windows. In the case of each alternate window the lower sashes open inwards on vertical axes, and in the center frame is fitted a lock, so arranged that the turning of the handle locks both windows. The two upper sashes fold downwards, and are guided on racks. These windows are considerably above the water line, and can, therefore, be opened even in the Channel. The lower deck is lighted by large circular port-holes, also fitted above the water line, so that no arti-

ficial light need be resorted to. There are many ventilating shafts from the deck above.

The promenade deck extends three-fourths of the ship's length, and is without obstruction, except the wheel-house and chart room, which form a center support for the captain's bridge. On this bridge are telegraphs, gongs and speaking tubes, enabling the captain to communicate orders to the engineer, and forward and aft, so that he has absolute control of the ship, and this, with the fore and aft rudder, makes it easy to steer the vessel stern first out of Boulogne between the long wooden jetties which protect the inner harbor.

The engines are compound, with two cylinders 56 and 110 inches diameter and 6 feet stroke, placed in an inclined position and working on a two-throw crankshaft. With the exception of the cylinders, which are of cast iron, the engines are entirely of forged and cast steel. The pistons have phosphor bronze packing with coach springs, while the piston valve to the high-pressure cylinder has Buckley's packing. The high-pressure cylinder piston valve is placed on the outside of the cylinder, and the slide valve of the low-pressure cylinder between the two cylinders, both operated by the usual double eccentric and link motion. The piston and connecting rods have a diameter of 10 inches. The crankshaft is in two pieces, which are interchangeable. The diameter is 20½ inches, as is also that of the paddle-wheel shafts, the connection being by flanged couplings. The paddle wheels have an over-all diameter of 22.54 feet, and the floats, which are curved, and on the feathering principle, are 13 feet long. They are constructed entirely of steel. The condenser is made of galvanized steel, and is cylindrical. It is placed athwartship under the engines, and between the cylinders and the cast-steel brackets for the shafting. The water is supplied to the condenser by a centrifugal pump, driven by two engines, each of which is capable of doing the maximum work. This pump is arranged to draw from the bilge in the event of a serious leak. The air pumps are driven by a lever connected by links to the low-pressure crosshead. There are two air pumps, as well as two bilge pumps and two sanitary pumps. To the

crankpins, crossheads, ahead guides, and main bearings the "Axiom" lubricator is fitted, grease being used, not oil, while the astern guides are fitted with stationary tubes on the same plan.

Two boilers are fitted forward and two abaft the engine compartments, so that the machinery is spread over three compartments, and occupies nearly one-third of the length of the vessel. The boilers are of the ordinary double-ended return-tube type, and are 14.5 feet in diameter by 18.75 feet long, each with six furnaces about 44 inches in diameter. There is a common combustion chamber to each pair of furnaces. The boilers are worked under a system of assisted draft, the stokeholds being open, with large ventilating shafts. There are also on board four double inlet fans 6 feet in diameter, driven by triple-expansion engines on the single-acting system. The boilers are fed by two of Weir's special engines. Fitted on the main and auxiliary pipes is a feed filter and feed heater. There are two separate smoke-pipes, the diameter being 10 feet, and the height from the furnace bars 60 feet. The boilers work under a pressure of 140 pounds to the square inch.

The vessel is thoroughly lighted by electricity. A vertical high-speed Robey engine with a 9½-inch cylinder and 9-inch stroke, having ordinary slide valve and a Pickering governor, is coupled direct to the armature of an ordinary inverted U-type dynamo, capable of running 360 lights at 250 revolutions, the electromotive force being 100 volts. The number of lights in the ship is 300. A special feature of the installation is the arrangement whereby any defect can be localized and joints repaired without inconvenience or damaging the woodwork of saloons, &c. The dynamo is in the engine room, and from it the electric current is led to a switchboard on the engineer's platform, whence the current is distributed to omnibus bars in various parts of the ship—on the lower deck aft, lower deck forward, main deck aft, main deck forward, upper saloon aft, and the machinery compartments and amidships. There is a fuse, and a switch on a slate base, and two wires led from the omnibus bar to each lamp, so that the only joints are at the lamp and at

the distributing centers. The navigating lights are electric, and on the bridge are two incandescent light projectors, with Edison-Swan double filament lamps for lighting the passenger gangways.

When tried on the Clyde, her mean speed for six runs on the mile was 21¼ knots, which was made with 56 revolutions of the engines, the indicated horse power being about 8,000.

Neptune and *Mercury*.—Two paddle-wheel steamers built by Napier, Shanks and Bell, for the Glasgow and Southwestern Railway Company, and engined by David Rowan and Son, Glasgow, were recently tried on the Clyde. Their dimensions are:

Length, feet, ...	220
Beam, feet, ...	26
Depth, feet, ...	9.5

The requirement for their trial was that they should maintain a speed of 17.75 knots for four hours; the *Neptune* made 18, and the *Mercury* 18.44. The speed of the latter was made with an air pressure of 1.3 inches of water in the fire room, the steam pressure being 115 pounds, the revolutions of the engines 57 per minute, and the indicated horse power about 2,000.

The engines are of the diagonal, compound, direct-acting type, with the cylinders placed side by side under the main deck. The cylinders work on two cranks set at an angle of 90 degrees. The high-pressure cylinder is 33 inches and the low-pressure cylinder 62 inches, both being adapted for a stroke of 5 feet. To insure the maximum of strength with the minimum of weight, steel and gun-metal have been used where practicable. The entablature for supporting the crankshaft, etc., is made of cast steel bolted to fore-and-aft cast-steel girders fastened down to the ship. The cylinders are bolted to these cast-steel girders, as well as to the engine seat, and the thrust of each piston is taken up by two main stays of ingot steel bolted to the main bearings at one end and to the cylinders at the other end. These stays also form guides for the piston rods. Each cylinder is fitted with a flat side valve worked by the usual double eccentric and link motion, and reversed by a direct-acting steam reversing engine placed in a con-

venient position below. The crankshaft is built of mild steel, and the paddle shafts and piston rods of the same material. A supply of circulating water is obtained from a gun-metal centrifugal circulating pump.

Steam is supplied by two boilers of the gunboat type. They are 9.5 feet in diameter by 18.5 feet long, designed for a working pressure of 115 pounds to the square inch. Each boiler has three furnaces, with Fox's corrugated flues. The arrangements admit of the boilers being worked under forced draft on the closed fire room system. Air is supplied to the furnaces by two large fans, each driven by an independent engine, each fan being capable of supplying the air necessary for the proper combustion of the fuel. The fans and engines are placed on the main deck beside the steam steering gear, which is forward of the engines; these fans and gear are always under the eye and within the control of the engineer on the starting platform.

They are provided with the usual independent feed and auxiliary pumps, and with evaporators for making up the loss of feed water for the boilers.

Special provision has been made for preventing loss of life in case of accident. The deck seats upon the promenade deck are fitted with water-tight copper buoyancy tanks below the seats, and provided with life-lines, and are so constructed that if thrown into the water each seat rights itself, and is certified by the Board of Trade to have a sufficient buoyancy to support twenty persons. In addition there are, of course, the statutory number of life jackets, lifebuoys, with two boats fitted with copper tanks, etc.

Slieve Bearnagh.—A paddle wheel steamer, similar to the preceding one, built by James and George Thomson, Clydebank, for the Belfast and County Down Railway Company, recently completed her trials with satisfactory results, making nearly 18 knots on the measured mile, and upwards of 17 for a continuous run of four hours. Her dimensions are:

Length, feet, ...	225
Breadth, moulded, feet, ...	26
over guards, feet, ...	54
Tonnage, gross, ...	383

She is of exceptional strength throughout. There are nine water-tight compartments, constructed in accordance with the recommendations of the Bulkhead Committee, and the vessel is practically unsinkable even with two adjacent spaces bilged. There is also a very complete outfit of buoyant deck seats, and other life-saving appliances. The engines are of the compound diagonal surface-condensing type, and are supplied with steam by a double-ended boiler of the ordinary marine type.

Proposed Steamers for the Canadian Atlantic Service.—Tenders were invited in June for the construction of four fast steamers for this service, but contracts have not yet been awarded. They are to be of approximately the following dimensions:

Length, over all, feet, ...	572
Beam, feet, ...	62
Depth, moulded, feet, ...	42
Draught, extreme, feet, ...	30

The engines will be of the twin-screw quadruple-expansion type, with four cranks each, and will be of about 21,000 horse power collectively, or of sufficient power to maintain a speed of 21 knots across the Atlantic. They will have bunkers for 3,000 tons of coal, and a cargo capacity for 3,500 tons, besides which there will be accommodations for 300 first-class, 200 second class and 1,000 steerage passengers.

Their construction is conditional on the granting of a subsidy of $750,000 a year by the Canadian Government.

Aberdeen.—A steamer built by Fleming and Ferguson for the Canadian government, and intended for use in the protection of the fisheries. She is 187 feet long, 31 feet beam and 19.25 feet deep. Her engines, which are of about 1,500 I.H.P., are of the quadruple-expansion type, with cylinders 18, 24, 35 and 54 inches diameter by 36 inches stroke, designed for a working pressure of 220 pounds per square inch. The most interesting feature of her construction, from an engineering point of view, is the fitting of water-tube boilers. She has two of the Fleming and Ferguson type, described and illustrated on page 358 of the last number of the JOURNAL.

Miowera.—A single-screw steamer, built by C. S. Swan and Hunter, Wallsend-on-Tyne, recently completed her trial, making a speed of 17 knots. She is 357 feet long, 42.25 beam and 28 deep. Her engines are 33, 53 and 85 inches diameter by 54 inches stroke, and are supplied with steam at 165 pounds pressure from four cylindrical boilers working on Howden's system of forced draft.

Caledonia.—A Peninsular and Oriental steamer, built by Caird and Co. Her dimensions are: Length, 486 feet; beam, 54 feet; depth, 37.5, and of 7,600 tons gross. She will have a single-screw, triple-expansion engines, with five cylinders, the I.H.P. of which will be 12,000. She will have a high rate of speed, and be superior in finish and appointments to any of the steamers of this fleet.

YACHTS.

Formosa.—A brief mention was made of this yacht in the last number of the JOURNAL, since which time she has been tried with satisfactory results. Her engine was built by the Atlantic Works, East Boston, and has cylinders 13½, 21 and 34 inches diameter by 22 inches stroke; the air pump is attached to the main engine, but the circulating pump is independent. She has two Almy boilers containing 62 square feet of grate and 2,394 of heating surface, arranged for working under forced draft on the closed fire room system. These two boilers measure 6.86 feet in length over casings, 13.75 in width, and 7.33 in height to the base of smoke-pipe hoods, and weigh 35,300 pounds empty and 38,500 pounds with water. The screw propeller is 7 feet 8 inches diameter and 9 feet 6 inches pitch.

The official trial of the yacht was of two hours duration under forced draft, burning anthracite coal with an air pressure of 1.62 inches of water in the fire room. Diagrams were taken from the cylinders of the main engine every ten minutes, and the average I.H.P. was 804, the main engine making 181.8 revolutions per minute. The revolutions of the blower engine were 700 per minute.

Zoraide.—A twin-screw steel yacht, built by Earle's Shipbuilding Company, Hull, England, from the designs of Mr. St. Clare J. Byrne. She has two masts with fore-and-aft sail only, and is of the following dimensions: Length on water line 162 feet; breadth, moulded, 27 feet; depth, moulded, 17.5 feet; yacht measurement 550 tons. Her engines are of the triple-expansion type, and work at a pressure of 175 pounds, steam being supplied by two cylindrical boilers. She has a square stern and a curved stem, a flush upper deck and topgallant forecastle, and a complete lower deck forward and aft of the boilers and engines. On the upper deck is a range of deckhouses containing a drawing-

room, vestibule and entrance to sleeping cabins, dining room, pantry, galleys, engine and boiler casings, and captain's room. The space on the lower deck aft is fitted up with the owner's sleeping cabins, bathrooms, etc., as also is the after part of the forward 'tween-decks. The passages, vestibules, dining-room and drawing-room are all panelled in solid oak and polished; and the officers' and crew's quarters are comfortably furnished in the forward 'tween-decks. A water ballast tank is fitted aft for trimming purposes.

Ellida.—A yacht built by Ramage and Ferguson, Leith, Scotland, for Mr. James Currie. She is 152 feet long, 22 feet beam, 13.5 feet deep, and 350 tons yacht measurement. Her engine is of about 600 I.H.P., and has cylinders 14, 22 and 36 inches diameter by 27 inches stroke, designed for a working pressure of 160 pounds. She has one large cylindrical boiler.

ERRATA.

The expression $e^{\frac{2\pi h}{\varepsilon V^2}}$ in equations 6, 7 and 8, on pages 263 and 264 of the last number of the JOURNAL, should be

$$e^{\varepsilon \overline{V^2}}$$

BOOK REVIEWS.

DICTIONARY OF ELECTRICAL WORDS, TERMS AND PHRASES.—BY PROFESSOR EDWIN J. HOUSTON. THIRD EDITION. New York: W. J. Johnston Co., 253 Broadway. Price, $5.00.

The third edition of this admirable work brings the subject fully up to the state of electrical science in 1894. The rapid growth of electrical engineering has brought into common use many words and phrases, the meaning of which is not thoroughly understood by many people who use them, which makes this dictionary of value to the general public, and of inestimable worth to the engineer, scholar and student. It is profusely illustrated, and the definitions, though concise, are accompanied with abundant examples, unabridged. Besides the electrical terms, it includes a sufficient number of definitions in the mechanic arts, and in mechanical and civil engineering to cover the entire field of electrical engineering.

The third edition is nearly one-fourth larger than the preceding one, and contains 582 illustrations. It is an octavo edition, double columns, bound in cloth, and presents a handsome appearance.

ELECTRICITY ONE HUNDRED YEARS AGO AND TO-DAY.—BY EDWIN J. HOUSTON, PH. D. New York: The W. J. Johnston Company, 253 Broadway. Price, $1 00.

This little book of 199 pages traces the history of electrical science from its birth to the present day, and is the result of much research on the part of the author. It is written in an agreeable style, much of the subject-matter having been embodied in a lecture delivered before the electrical section of the Brooklyn Institute, to which have been added copious foot-notes giving more detailed information regarding the particular discovery referred to. Aside from the value of the historical matter contained in the book, there is much information to be derived from it.